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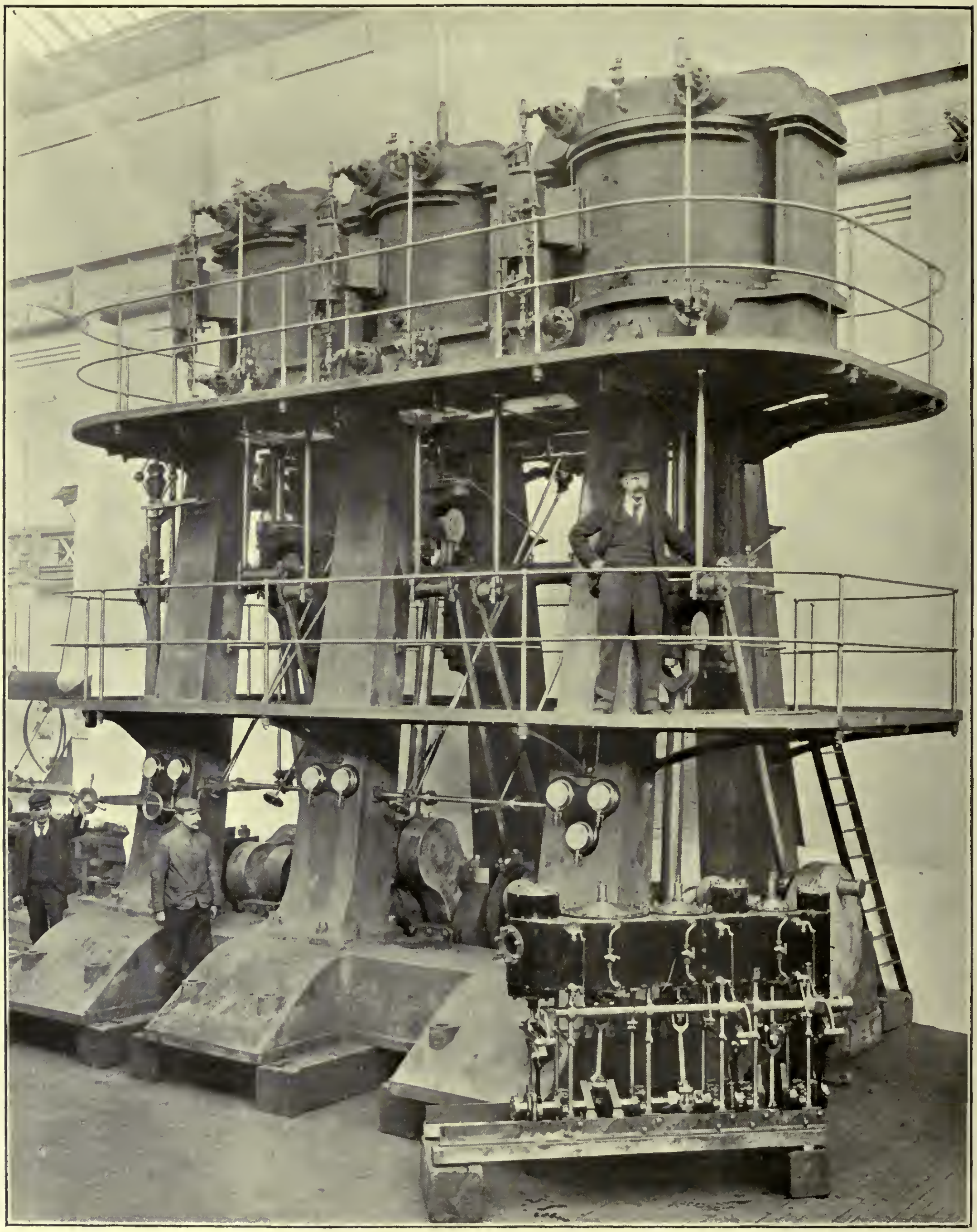
MODERN ENGINES
AND
POWER GENERATORS



Frontispiece.]

DIGNITY AND IMPUDENCE.
TWO ENGINES OF EQUAL POWER.

MODERN ENGINES

AND

POWER GENERATORS

A PRACTICAL WORK ON PRIME MOVERS AND THE TRANSMISSION OF POWER STEAM, ELECTRIC, WATER, AND HOT AIR

BY

RANKIN KENNEDY, C.E.

AUTHOR OF

"ELECTRICAL INSTALLATIONS" "ELECTRICAL DISTRIBUTION BY ALTERNATING CURRENTS AND TRANSFORMERS" "PHOTOGRAPHIC AND OPTICAL ELECTRIC LAMPS" AND NUMEROUS SCIENTIFIC ARTICLES AND PAPERS ON MECHANICAL AND ELECTRICAL ENGINEERING

WITH MANY HUNDRED ILLUSTRATIONS

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PREFACE TO VOLUME IV

THIS Volume deals principally with the Reciprocating Steam Engine and its many accessories. This prime mover is by far the most important engine at present, but there are unmistakable signs that it has reached a point in its progress beyond which improvement is impossible. The adoption of superheating in order to increase the economy of fuel is the last step, beyond which it is difficult to see in what other direction further improvements may come from. The steam turbine has already proved a superior prime mover in marine propulsion for the higher classes of high-speed vessels. The gas-blowing engines have displaced the reciprocating steam engine at the great iron works here and abroad, and the gas engine with the gas producer is more and more taking the place of steam engines.

The small factory with its tall chimney and boiler and engine is already out of date; such factories being established now would be driven by oil or gas engines, or by an electric motor on the street electricity supply. Nevertheless there is, and will be for long, a considerable demand for reciprocating steam engines. It will take many years to bring turbines and gas engines into universal application, so that the study of reciprocating steam engines and their modern development is still of great interest and value. Thanks are due to the many engineers who have supplied information and drawings; to Mr. Morison, of Messrs. Westgarth & Co., and to Professor Watkinson, for details of their inventions.

RANKIN KENNEDY.

GLASGOW, *November* 23, 1904.

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MODERN ENGINES AND POWER GENERATORS

CHAPTER I

THE STEAM ENGINE

INTRODUCTORY

THIS the first, and for many decades almost the only, prime mover of great power, still holds the foremost place, and is likely to do so for at least another decade. The internal combustion engine is, however, already pressing the competition hard against the steam engine. The steam engine, when it was first introduced, had no competitor, and the conditions under which it worked were favourable. Coal and water were plentiful and cheap, and so was land. Engineers at their leisure improved it in detail; there was no hustle or hurry in the improvement of steam engines. They remained for half a century, at least, pretty much as James Watt left them. Even twenty-five years ago the problems of steady turning and governing at constant speed with variable loads were understood by a very small percentage of steam engineers. Rough and ready practice and rule-of-thumb design was characteristic of steam engine work. Slow speeds and huge engines were firmly believed in; great massive beam engines and immense gear wheels were quite commonly in use, giving out comparatively small powers. These engines were admired more for their immense size than for the actual power they developed.

The locomotive and marine engineer, and after them the electrical engineer, compelled the steam engine designers to admit improvements. The marine engineer insisted upon high economy of fuel and great power output from small engines; and high-speed gearing he would not have on any account. And similarly to the locomotive engineer, small engines at high speeds and high powers per pound of weight were his requisites; but in his case economy of fuel was not so important as it was with the marine engineer. The locomotive engineer and the marine engineer, however, did not need to trouble themselves about constancy of speed, or even turning movements: governors to them were quite superfluous.

Large land engines of slow speeds were governed by pendulum governors of Watt's type in most cases. The variation in load on these engines, however, was never very great, for in large textile and other factories in which they were used the constant load due to the transmission and distribution of power plant was never less than 75 per cent. of the full load, so that the governor was, as a rule, more ornamental than useful. It might, once in fifty years, be of use in case a main shaft broke, when it would perhaps prevent the engine from racing, and the fly-wheel from bursting.

Porter, an American, introduced, through Sir J. Whitworth, the first high-speed well-governed engines from America. From the same country came the Corliss valve gear, with automatic variable expansion governor and cut-off.

Advanced engineers saw at once the advantages of high speeds and perfect governing when electrical dynamo machines came on the scene thirty years ago. The electric light revealed every variation in speed and turning movement in the engine driving the dynamo. The electrical engineer's business was absolutely ruined if the engine did not govern within 3 per cent. between no load and full load. He demanded perfect governing with high speed and the highest economy,—three requisites for his business which were, at the date of the introduction of the first dynamos, very rarely combined in one engine. It has been found of importance that not only should the engine make a regular number of strokes per minute, but also that the angular velocity of the fly-wheel should be constant. This requirement is one demanded by alternators, and is theoretically impossible of attainment with a single reciprocating engine, the impulses being alternated between two dead centres. By combining three engines on cranks at 120° apart, regular angular velocity may be attained.

In the matter of steam economy, little progress has been or can be made in modern steam engines. The economy depends upon high pressure of steam, superheating, proper automatic cut-off and expansion in the cylinder, and a good vacuum in the condenser. The introduction of mild steel plates, and heavy boiler-making tools of great precision, have enabled the boiler maker to turn out boilers for pressures up to 250 lbs. per square inch, with shells 10 to 12 feet diameter. Higher pressures may be used, but it is doubtful if the gain will compensate for increased cost of maintenance and repairs.

About superheating steam for steam engine working there are differences of opinion, which require settlement by direct tests as to the degree of superheating most economical in practice; while there is no difference of opinion, or experience, as to the fact of the beneficial effects of a small degree of superheat sufficient to prevent cylinder and pipe condensation.

Superheating steam in pipes or a vessel open to the boiler expands the steam at the constant boiler pressure, and a small amount of heat produces considerable expansion.

The heat required to generate 1 lb. of dry saturated steam is about 1200 ⊖; to superheat 1 lb. of steam to 150° Fahr. requires 72 ⊖. And this increase of temperature expands the volume 30 per cent.; that is, for 6 per cent. of heat added we get 30 per cent. increase of volume. So that if admitted to a cylinder for the same time of admission as ordinary steam, the weight of steam used would be 30 per cent. less.

Where superheating is carried only far enough to annul condensation in pipes and cylinders the increase of volume is of no advantage, for the steam becomes saturated before cut-off, and condensation occurs during expansion.

In practice, superheating to 150° is found practicable without injury to valves or cylinders, and in large engines even 200° can be used. This increased heat is, however, nearly all lost before or at the point of cut-off; and condensation occurs during the rest of the stroke. The water formed assists the lubrication.

With these moderate degrees of superheat, good mineral cylinder oil can be used.

Using ordinary steam, even under modern conditions, between 12 to 30 per cent. of the steam is condensed between the boiler and the engine cylinder during its admission.

Steam itself does not readily leak past valves or pistons in any great quantity, but when it condenses it, in the form of hot water, leaks to an extent much greater than is usually imagined—the volume of water weighing much more than the same volume of steam. Where water condenses on the surfaces the leakage may be sixty times as great as it would be with dry superheated steam.

Superheated steam behaves as a permanent gas, so that it flows steadily through nozzles and passages of turbines. Saturated steam, in flowing through a nozzle such as De Laval's, produces a rhythmic sound; this is due to condensed water periodically choking the throat of the nozzle. With a slight superheat, the sound becomes a steady hiss like that produced by escaping air. By the way, this hissing sound of escaping fluids under pressure indicates some intermittent effect of great frequency, but it has never been properly accounted for.

Dry steam in steam turbines is of great importance, for the water present in ordinary steam clogs the spaces between wheels and guides and produces friction.

In a pressure turbine superheating has most beneficial effect, but in an impulse turbine its effect is somewhat decreased by the increase of volume and reduction of weight of the steam. The impulse being due to $\frac{WV}{g}$, it is evident a reduction in the value of W, the weight of steam, will reduce the impulse.

Other methods for reducing steam condensation may be enumerated :—

1. Compounding.
2. Steam jacketing.
3. Reduction of clearance surface and volume.

But superheating is by far the most economical and efficient method.

The compounding of engines and the boiler pressures are now at their highest limits, so far as we can see, so that the reciprocating engine can only be further improved by superheating.

Another possible direction for improvement lies in the compounding of reciprocating engines with steam turbines. The turbine gives a far better return for low-pressure steam and a good vacuum than the reciprocating engine; while, on the other hand, the reciprocating engine gives a better result with the high pressure-steam. This points to the combination of high-pressure reciprocating with low-pressure turbine engines.

Again, compounding and superheating may be combined: by providing superheaters between each engine in a compound engine, a superheater between the high-pressure and intermediate, and another between the intermediate and low-pressure cylinders. This idea is not new, although it has been assumed to be so only recently. The frightful complications introduced by all the auxiliary machinery introduced to increase the efficiency of steam deterred the old engineers from using them. The younger engineers having nothing left to improve upon mechanically in engines, are inclined to adopt these expedients, and take the risk of the complications; but the result is too often a slight gain in efficiency with a great loss in simplicity and durability.

The municipal electric light engineer has provided the world with stations which are veritable museums of old ideas in modern dress. If any one wants to see how many auxiliary machines can be hung on to the modern steam prime mover, let him visit an electric light station. A simple steam prime mover consists of a steam boiler and engine only, with a few simple machines such as a feed pump, pressure and water gauges.

To this simple combination we may with advantage and little cost add an exhaust steam feed water heater, thus utilising some of the rejected heat in the steam. But if water in plenty is available and cheap, instead of an exhaust feed water heater utilising only a part of the rejected heat, we may add a condenser and obtain another 14 lbs. difference of pressure on the two sides of the piston. Having in this way disposed of the exhaust steam, experts on boilers will insist on having hot feed water, so that the adoption of condensing leads to adopting a feed water heater of some kind, generally a series of pipes placed in the current of hot gases between the boiler furnaces and the chimney, an arrangement called an "economiser."

The nature of the condenser will depend upon the water supply. It may be a simple affair where plenty of water with a fall of 16 to 20 feet is available; it may then be a simple jet or ejector condenser. But as water in plenty with a fall of 16 to 20 feet is generally a valuable commodity, and in most places scarce, the condenser is usually of the more complicated and expensive types, with air pumps, and requiring motive power for the air pump, and also for pumping up the condensing water; so that it becomes a question whether the condensing is worth the additional cost, maintenance, and attendance. In addition to a condenser with an air pump and circulating water pump, there are in electrical stations in many cases huge arrangements made for cooling the condensed water, so that the same water can be used over again. These cooling apparatus generally require additional motive power to pump the hot water through them, so that what was at first only a boiler and engine may very soon develop into quite a complicated conglomeration in our attempts to capture some of the rejected heat in the exhaust steam. The question arises, just how much is really gained by using a condenser with two pumps and an economiser, when compared with a simple exhaust steam feed water heater; and also how much is gained when, in addition to these, we use water-cooling plant.

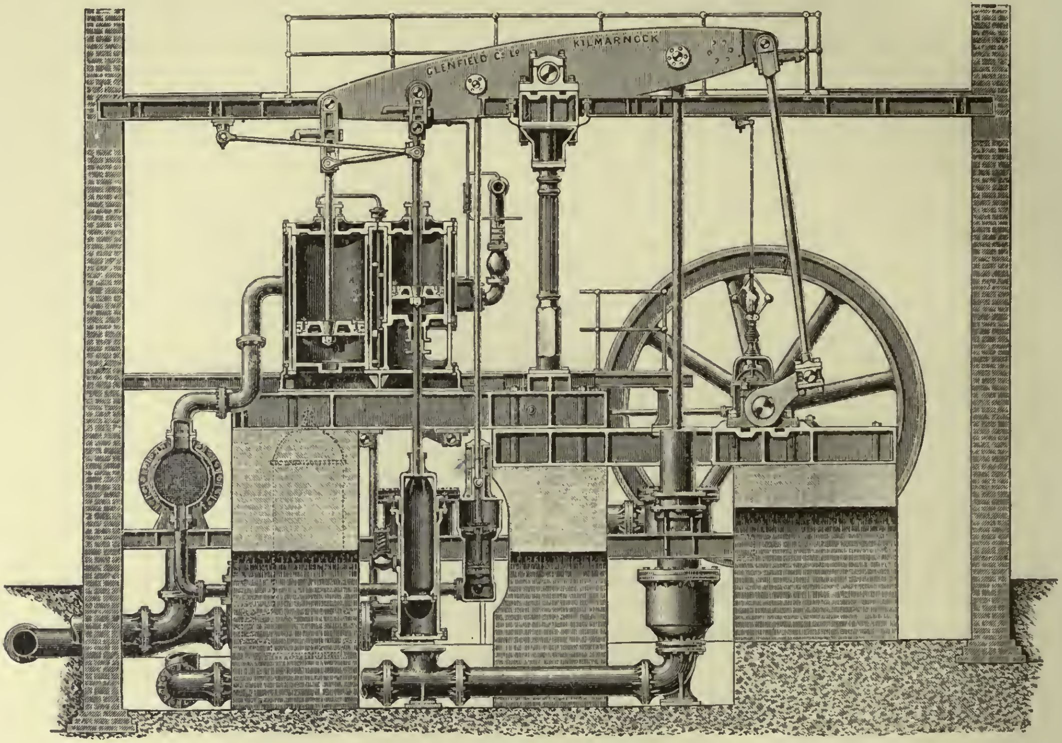


FIG. 1.—Pumping Beam Engine.

In some cases it is necessary to have a water softening plant, to preserve the boilers from some soluble salts in the water.

Then in most cases separators are employed in the steam pipes between the boilers and engines to separate the water from the steam before it enters the engine. And in other instances an oil separator is required to separate the oil from the condenser water before it is again pumped into the boiler.

Another addition is often made in the shape of a fan or blower for forcing the draught of air into the boiler furnaces; and mechanical stokers may be added to feed in the fuel.

All these adjuncts to the steam engine and boiler are in themselves admirable contrivances, but their adoption in any given case is a matter for the careful consideration of the experienced engineer, who can scientifically weigh up the account for and against their use in any given circumstances.

There are, again, the questions arising from cylinder condensation, such as steam jacketing, compounding, and superheating, all of which add to the cost of maintenance and attendance.

The marine engine—situated as it is in the midst of plenty of cold water only costing the slight power required for the circulation, the load nearly always constant and full power, the engine situated in close proximity to the boilers—may be taken as a steam prime mover at its best in the best of circumstances. In a modern steamship will be found all the auxiliaries and appendages required to get the most out of the fuel in the shape of brake horse-power on the propeller shaft.

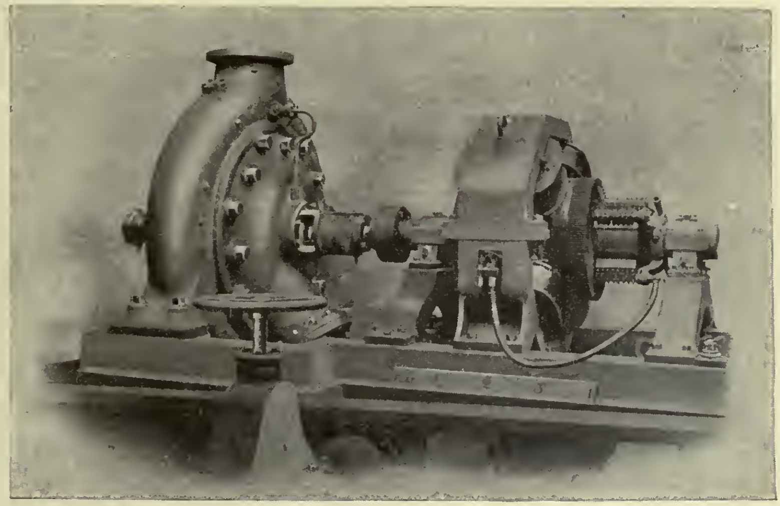


Fig. 2.—Pumps and Motor.

It is in this necessity for many additional contrivances, to enable the steam prime mover to reach its present-day economy, wherein its failure to compete with the internal combustion engine arises.

The day of the small steam engine is past already. The forest of boiler chimneys in manufacturing towns will never be rebuilt, and many of them are not now in use, while a great many more should with economy and many advantages be abandoned. The oil and gas engine, and the electromotor, can furnish power for manufacturing purposes up to at least 250 horse-power cheaper than any local steam engine plant.

Small steam engines are still in use on board steamers for many purposes, but that is only because of the inertia and conservatism of marine engineers. All these small engines are a source of waste and trouble, and should be replaced by electromotors driven from a central dynamo in the engine-room.

The steam turbine is the final stage of the steam engine's progress. While, however, it has reduced the complications, weight, and bulk of the engine, it still requires all the additional auxiliary plant to obtain economy.

It would be useless to relate the well-known history of the reciprocating steam engine. We have only to deal with its modern forms useful at the present day. As an example of the survival of an old type of prime mover adopted for a particular purpose, Fig. 1 is interesting. The high and low-pressure cylinder and one pump are seen under the left-hand end of the beam; another pump and fly-wheel are at the other end. This type of engine for long was a general favourite among old time engineers. Much the same type of engine may be seen at work still, and even afloat; they are in use largely about New York waters, propelling ferry and pleasure steamers.

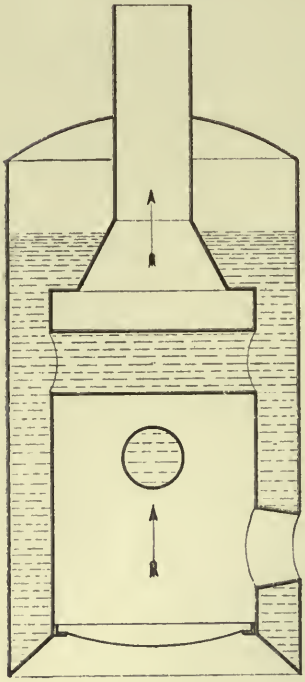


FIG. 3.—Cross Tube Boiler.

The beam engine shown with two plungers lifts 746 gallons per minute 290 feet with 70 lbs. pressure of steam, giving a pump horse-power of 66.

Comparing this with a modern pumping plant, in which the pumps are high-lift centrifugals, and the power transmitted from the engine electrically, as shown in Fig. 2, this plant lifts 2800 gallons per minute to a height of 265 feet. The motor is 345 brake horse-power.

The same pumps are made to lift to 800 feet and 3500 gallons per minute, and may, of course, be coupled to steam turbines,—so that the whole plant is rotary and valveless, guideless, slideless, and rodless.

This strikingly illustrates the trend of modern improvements: simplification is the order of the day.

Boilers for steam engines shall be taken in place further on. Meanwhile we may refer to the elementary types in use.

Fig. 3 is a sectional view of the common cross tube vertical boiler, a type which has been elaborated into large and powerful tubular boilers; but it is mostly used for small and portable powers. Fig. 4 is a cross section of the common Lancashire boiler. It has two flues. When made with one flue it is called a Cornish boiler. As a steam boiler it is hard to beat. But it requires a good deal of room for setting it up with the necessary flues around; hence is not always applicable.

Boilers for large powers in small space are designed with many tubes, so that large flues are not required, and great heating surfaces are presented in small bulk.

The return tube dry back boiler, shown in Fig. 5, is a type in common use. Fig. 6 shows it with a wet back, not so common; for it is better not to cool the gases before they enter the tubes.

Generally speaking, tubular boilers require a more powerful air draught than the Lancashire boiler. The common locomotive boiler shown in Fig. 7 is a tubular boiler. The draught is, in the locomotive, provided by the blast of the exhaust steam—the steam blast being one of Stephenson's improvements on his locomotive. Morton, in his ejector condenser patents, further improved the blast, to produce a partial vacuum in the locomotive cylinders, by working on the ejector principle. (See Vol. I. p. 102.) It is shown in Fig. 8. The λ-shaped pipes are the exhaust pipes. The cranks of the engine are at right angles, so that one jet of steam is in full blast when the other is just ending.

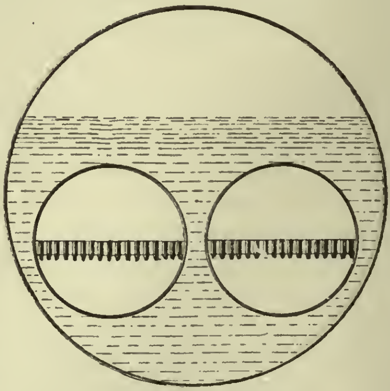


FIG. 4.—Lancashire Boiler.

There are many types of tubular boilers—some good, some indifferent, some bad.

The British naval engineers responsible for the selection of boilers preferred the later kind when they adopted a tubular boiler some years ago for war vessels.

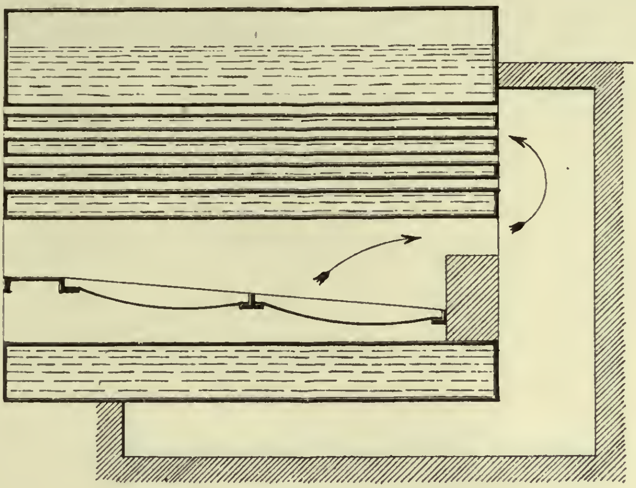


Fig. 5.—Tubular Dry Back Boiler.

The boiling of water in a closed boiler to produce steam is not such a simple matter as one would suppose. Water, like air, is a bad conductor of heat. It can be heated quickly only by convection; that is, by the heated water rising from the hot surface of the boiler shell or tube to make room for the cooler water.

The Babcock & Wilcox boiler is an example of a good, well-designed tubular boiler, shown partly in section in Fig. 9. Mr. George H. Babcock, in his lectures at Cornell University, very clearly and simply stated the elementary science of steam making, and we quote from these lectures to put his views and the facts briefly. If we take a quantity of ice, say 1 lb., at absolute zero and supply heat, the first effect is to raise its temperature until it arrives at a point 492° Fahr. above the starting-point. Here it stops growing warmer, though we keep on adding heat. It, however, changes from ice to water, and when we have added sufficient heat to have made it, had it remained ice, 283° hotter, or a temperature of 315° by Fahrenheit's thermometer, it has all become water, at the same temperature at which it commenced to change, namely, 492° above absolute zero, or 32° by Fahrenheit's scale. Let us still continue to add heat, and it will now grow warmer again, though at a slower rate,—that is, it now takes about double the quantity of heat to raise the pound 1° than it did before,—until it reaches a temperature of 212° Fahr., or 672° absolute (assuming that we are at the level of the sea). Here we find another critical point. However much more heat we may apply, the water, as water, at that pressure, cannot be heated any hotter, but changes on the addition of heat to steam; and it is not until we have added heat enough to have raised the temperature of the water 966°, or to 1178° by Fahrenheit's thermometer (presuming for the moment that its specific heat has not changed since it became water), that it has all become steam, which steam, nevertheless, is at the temperature of 212°, at which the water began to change. Thus over four-fifths of the heat which has been added to the water has disappeared or become insensible in the steam to any of our instruments.

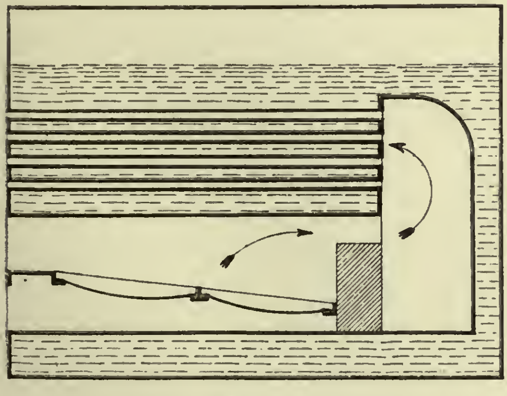


Fig. 6.—Wet Back Boiler.

It follows that if we could reduce steam at atmospheric pressure to water, without loss of heat, the heat stored within it would cause the water to be *red hot*; and if we could further change it to a solid, like ice, without loss of heat, the solid would be white hot, or hotter than melted steel—it being assumed, of course, that the specific heat of the water and ice remain normal, or the same as they respectively are at the freezing-point.

After steam has been formed a further addition of heat increases the temperature again at a much faster ratio to the quantity of heat added, which ratio also varies according as we maintain a constant pressure or a constant volume.

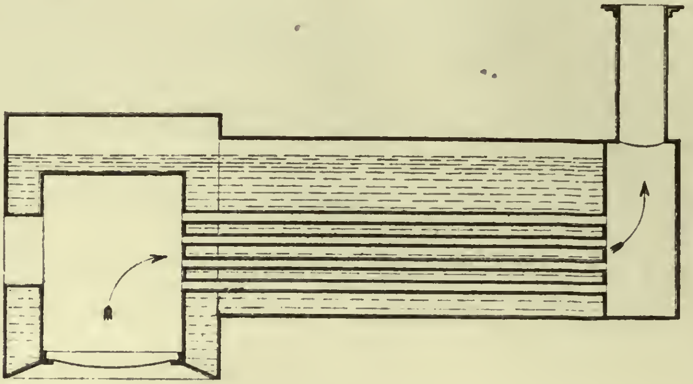


Fig. 7.—Locomotive Boiler.

The heat which has been absorbed by 1 lb. of water to convert it into a pound of steam at atmospheric pressure is sufficient to have melted 3 lbs. of steel or 13 lbs. of gold. This has been transformed into something besides heat,—stored up to reappear as heat when the process is reversed. That condition is what we are pleased to call latent heat, and in it resides mainly the ability of the steam to do work.

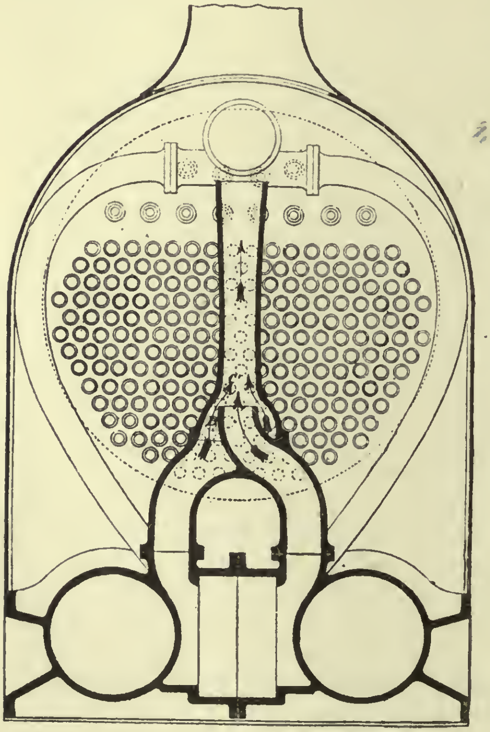


Fig. 8.—Locomotive Blast.

In order to generate steam, then, only two steps are required: first, procure the heat; and second, transfer it to the water. Now, we have it laid down as an axiom, that when a body has been transferred or transformed from one place or state into another, the same work has been done and the same energy expended, whatever may have been the intermediate steps or conditions, or whatever the apparatus. Therefore, when a given quantity of water at a given temperature has been made into steam at a given temperature, a certain definite work has been done, and a certain amount of energy expended, whatever the heat may have been obtained, or whatever boiler may have been employed for the purpose.

A pound of coal or any other fuel has a definite heat-producing capacity, and is capable of evaporating a definite quantity of water under given conditions. That is the limit beyond which even perfection cannot go, and yet we have known of cases where inventors have claimed, and so-called engineers have certified to, much higher results.

The first step in generating steam is in burning the fuel to the best advantage. A pound of carbon will generate 14,500 British thermal units (⊖) during combustion into carbonic dioxide, and this will be the same whatever the temperature or the rapidity at which the combustion may take place. If possible, we might oxidise it at as slow a rate as that with which iron rusts or wood rots in the open air, or we might burn it with the rapidity of gunpowder, a ton in a second, yet the total heat generated would be precisely the same. Again, we may keep the temperature down to the lowest point at which combustion can take place, by bringing large bodies of air in contact with it, or otherwise, or we may supply it with just the right quantity of pure oxygen, and burn it at a temperature approaching that of dissociation, and still the heat units given off will be neither more nor less. It follows, therefore, that great latitude in the manner or rapidity of combustion may be taken without affecting the quantity of heat generated.

But in practice it is found that other considerations limit this latitude, and that there are certain conditions necessary in order to get the most *available* heat from a pound of coal. There are three ways, and only three, in which the heat developed by the combustion of coal in a steam boiler furnace may be expended.

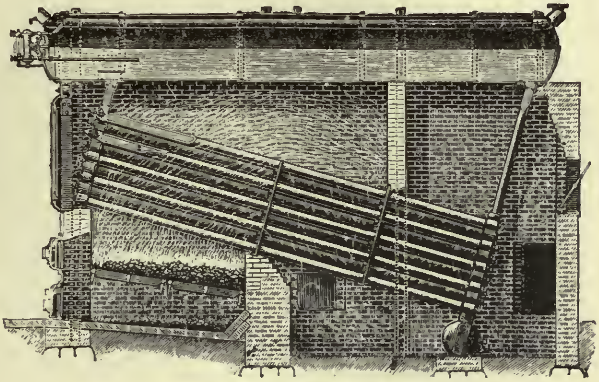


FIG. 9.—Babcock & Wilcox Boiler.

First, and principally, it should be conveyed to the water in the boiler, and be utilised in the production of steam. To be perfect, a boiler should so utilise all the heat of combustion; but there are no perfect boilers.

Second, a portion of the heat of combustion is conveyed up the chimney in the waste gases. This is in proportion to the weight of the gases, and the difference between their temperature and that of the air and coal before they entered the fire.

Third, another portion is dissipated by radiation from the sides of the furnace. In a stove the heat is all used in these latter two ways—either it goes off through the chimney or is radiated into the surrounding space. It is one of the principal problems of boiler engineering to render the amount of heat thus lost as small as possible.

The loss from variation is in proportion to the amount of surface—its nature, its temperature, and the time it is exposed. This loss can be almost entirely eliminated by thick walls and a smooth white or polished surface, but its amount is ordinarily so small that these extraordinary precautions do not pay in practice.

It is evident that the temperature of the escaping gases cannot be brought below that of the absorbing surfaces. This is supposing that all of the escaping gases have passed through the fire. In case air is allowed to leak into the flues and mingle with the gases after they have left the heating surfaces, the temperature may be brought down to almost any point above that of the atmosphere, but without any reduction in the amount of heat wasted. It is in this way that those low chimney temperatures are sometimes attained which pass for proof of economy with the unobserving. All surplus air admitted to the fire, or to the gases before they leave the heating surfaces, increases the losses.

We are now prepared to see why and how the temperature and the rapidity of combustion in the boiler furnace affect the economy, and that, though the amount of heat developed may be the same, the heat available for the generation of steam may be much less with one rate or temperature of combustion than another.

Water is practically a non-conductor of heat, though not entirely so. If ice-cold water was kept boiling at the surface the heat would not penetrate sufficiently to begin melting ice at a depth of 3 inches in less than about two hours. As, therefore, the heated water cannot impart its heat to its neighbouring particles, it remains expanded, and rises by its levity, while colder portions come to be heated in turn, thus setting up currents in the fluid. This is heating by convection.

Now, when all the water has been heated to the boiling-point corresponding to the pressure to which it is subjected, each added unit of heat converts a portion, about seven grains in weight, into vapour, greatly increasing its volume; and the mingled steam and water rise more rapidly still, producing ebullition such as we have noticed in the kettle. So long as the quantity of heat added to the contents of the kettle continues practically constant, the conditions remain similar to those we noticed at first—a tumultuous lifting of the water around the edges, flowing toward the centre and thence downward; if, however, the fire be quickened, the upward currents interfere with the downward, and the kettle boils over (Fig. 10).

If now we put into the kettle a vessel somewhat smaller (Fig. 11) with a hole in the bottom, and supported at a proper distance from the side so as to separate the upward from the downward currents, we can force the fires to a very much greater extent without causing the kettle to boil over, and when we place a deflecting plate so as to guide the rising column toward the centre, it will be almost impossible to produce that effect. This is the invention of Perkins in 1831, and forms the basis of very many of the arrangements for producing free circulation of the water in boilers which have been made since that time. It consists in dividing the currents so that they will not interfere each with the other.

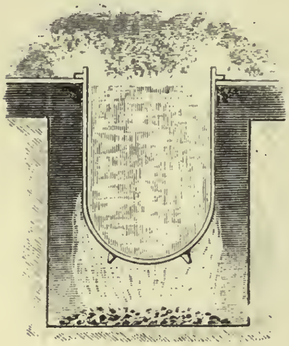


FIG. 10.—Boiling Water.

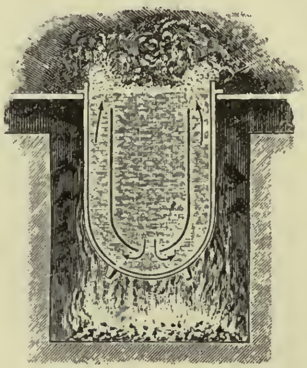


FIG. 11.—Boiling Water.

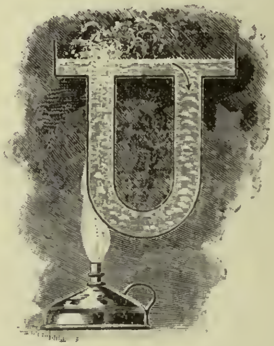


FIG. 12.—Water-Tube Boiler.

If now we take a U-tube depending from a vessel of water (Fig. 12) and apply a lamp to one leg, a circulation is at once set up within it, and no such spasmodic action can be produced. This U-tube is the representative of the true method of circulation within a water-tube boiler, and the elementary water-tube boiler.

The Babcock & Wilcox water-tube boiler has all the elements of safety, in connection with its other characteristics of economy, durability, accessibility, etc. Being composed of wrought-iron or steel tubes, and a drum of comparatively small diameter, it has a great excess of strength over any pressure which it is desirable to use. As the rapid circulation of the water ensures equal temperature in all parts, the strains due to unequal expansion cannot occur to deteriorate its strength. The construction of the boiler, moreover, is such that, should unequal expansion occur under extraordinary circumstances, no objectionable strain can be caused thereby, ample elasticity being provided for that purpose in the method of construction.

In this boiler, so powerful is the circulation that as long as there is sufficient water to about three parts fill the tubes, a rapid current flows through the whole boiler; but if the tubes should finally get almost empty, the circulation then ceases, and the boiler might burn and give out; by that time, however, it is so nearly empty as to be incapable of great harm if burst.

There are other equally good tubular boilers, both for land and marine purposes, to be described fully under "Boilers."

The engine may now be briefly reviewed in its simpler forms. The engine shown in Fig. 13 is one of the modern types of small power engines, a vertical inverted cylinder engine with a throttle governor for fairly high speeds.

The governor is of the fly-weeel inertia type.

This type possesses many improved features of construction. The governor is extremely sensitive, and suitable for electric lighting. The engine is fitted with a central oil box for continuous lubricating of the working parts, thus enabling runs to be made for long periods without stopping.

These engines can be worked condensing, where an independent condensing plant is available, in which case the same power can be obtained with 10 lbs. lower pressure. Condensers are not combined with these engines, on account of their comparatively small size.

The cylinder is fitted with a piston valve, which works in a renewable liner, and has also provided relief valves and bosses fitted with brass plugs for indicators. The lagging is of sheet steel, secured with brass screws.

The main bearings, large end connecting rod, and valve spindle guide bracket are all lined with anti-friction metal. The wearing parts are adjustable where required, and are all particularly large, so that on long runs there is no fear of heating.

The standard is self-contained, and carries in one casting the bored guides and main bearings. The top is faced, and bored to receive stuffing box and cylinder.

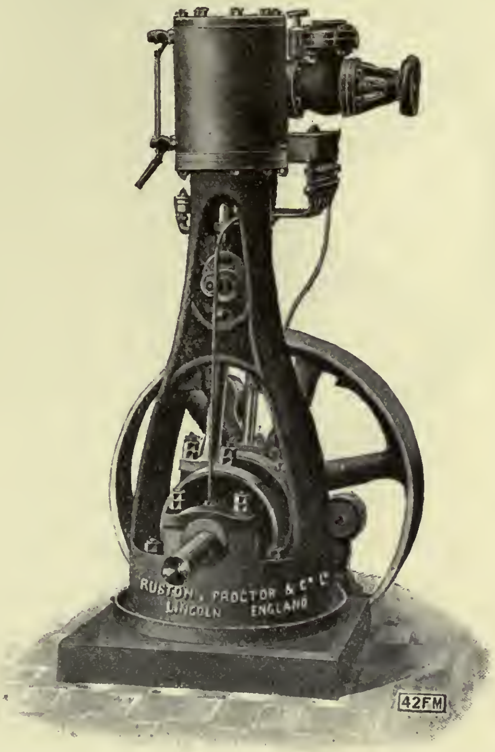


Fig. 13.—Vertical Engine.

The crank shaft is of mild steel, and fitted with circular cast-iron balancing weights, turned on the face and edge. It is long enough to receive a pulley on the side opposite to the fly-wheel, or a half coupling can be fitted for bolting direct to a dynamo.

The connecting rod is made of forged steel, with a marine pattern large end and solid at the small end, the bearings of both ends being adjustable.

The cross-head is of malleable iron, and fitted with large cast-iron slippers, which can be adjusted for wear.

The lubricating arrangements consist of an oil box and separate pipes conducting the oil to each moving part. This oil box is fitted with adjustable sight drop taps for continuous lubrication.

The improved fly-wheel inertia governor is fitted with two weights, one of which acts by its inertia and the other responds to centrifugal force. The latter is arranged as a box, fitted with several discs secured by a screw. By taking away a number of these discs the speed of the engine may be varied from 300 to 350 revolutions per minute.

The governor controls the speed by automatically altering the position of the eccentric, and thereby varying the point of cut-off of the steam in the cylinder.

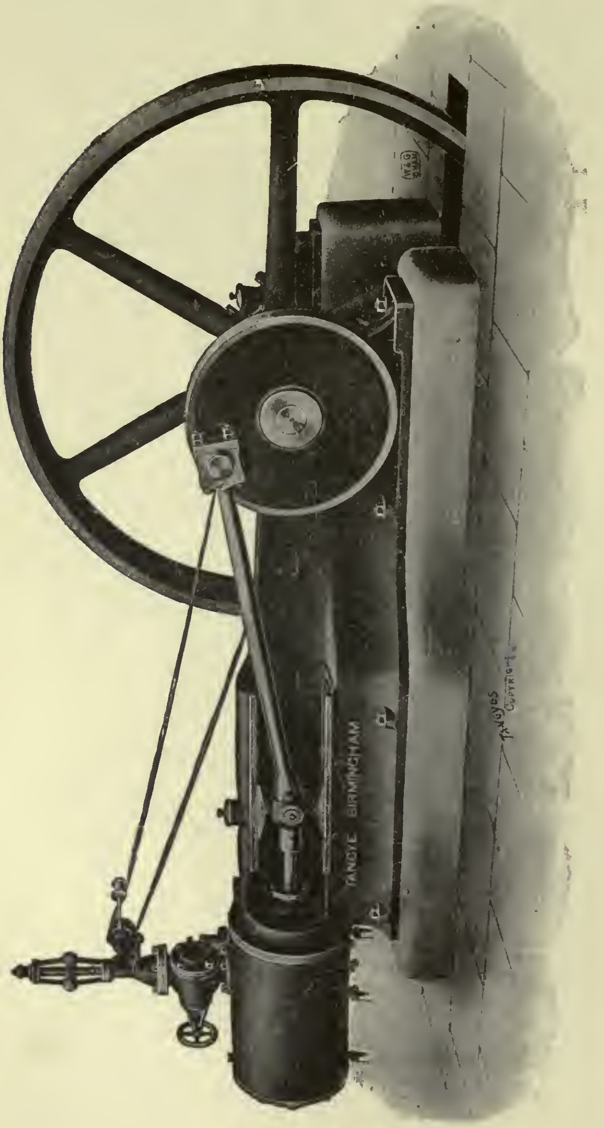


Fig. 14.—Horizontal Engine.

Another common type of steam engine of simple form is shown in Fig. 14, a horizontal engine with a throttle centrifugal governor. The modern form of bed-plate and slides are here very well illustrated.

These two engines were at one time the most popular small prime movers, and as steam engines were exceedingly handy and easily kept in repair. Another type largely used is shown in Figs. 15 and 16 in section and in Fig. 17 in complete view. This is a single-acting 2-cylinder engine with a crank case.

From the sectional views it will be further seen that the internal moving parts consist of two pistons, two connecting rods, crank shaft, single central valve, eccentric rod for same, and an automatic expansion governor.

The crank case is kept half filled with oil and water, which, when the engine is running, is in a few minutes churned into a foam and, while it offers little or no resistance to the moving parts, thoroughly lubricates all internal bearings constantly and, by using the same oil over and over again, economically. A sight-feed lubricator for the steam pipe is provided, and this is the only part of the engine requiring daily attention. A little oil may at stated periods be poured into the crank case by the vapour pipe or oil funnel, which also allows any vapour which may in course of time escape from the bottom of the pistons free exit to the atmosphere. The depth of oil and water in the crank case is limited by the height of overflow of the oil separator. This contrivance also, by allowing water only to escape, further economises oil. The water level in the crank case should never be allowed to fall more than 1 inch below the outlet of the separator, and the best oil to use in making up level is "Black Virginia," costing about 1s. per gallon. Expensive light oils must on no account be used, as they will not mix with water.

The steam is distributed to the cylinders by a single valve controlling supply and exhaust, and acts only on the top of the pistons. This is a very important and, indeed,

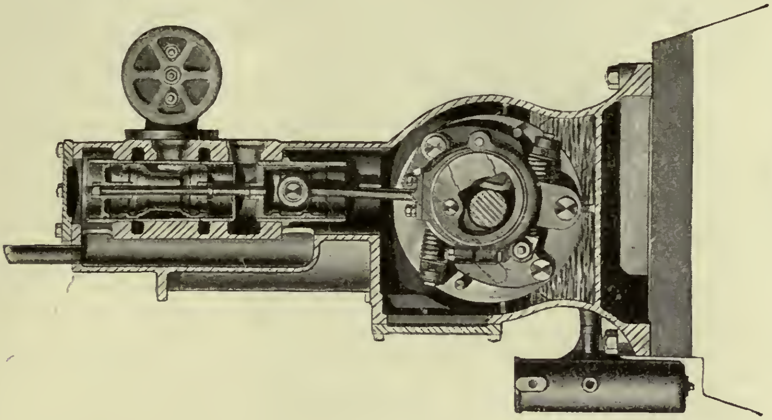


Fig. 16.—Transverse Section.

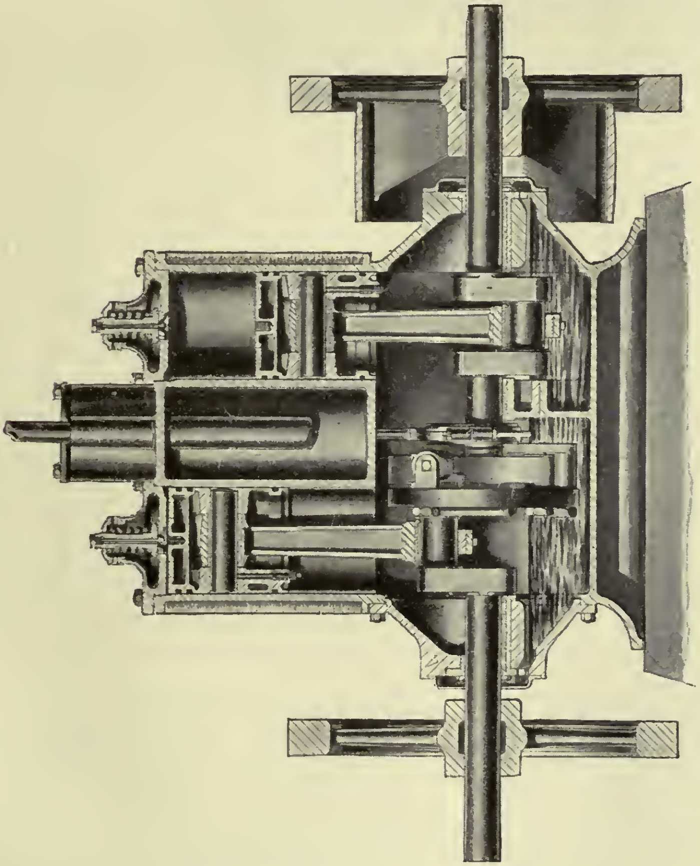


Fig. 15.—Longitudinal Section.

Single-acting Engine.

an essential feature in high-speed engines of this type, as it enables the connecting rods to be kept always in compression, and permits of the engine working silently even though the connecting rod and crank shaft brasses are worn $\frac{1}{8}$ inch slack. It is well understood that with the ordinary forms of double-acting engines the direction and nature of stresses are changed twice in each revolution from compression to tension, and *vice versâ*, as the steam is alternately admitted to each side of the piston, so that, unless the engine is provided with a system of forced lubrication under pressure, there is the necessity of continually taking up the slackness due to wear and knocking. With the single-acting engine, however, the pressure of the steam on the piston always acts in a downward direction throughout the whole revolution, so that the connecting rod is in constant compression and the pressure on the main bearings always on the bottom half bush; hence, no matter how much lost motion exists, it is always automatically taken up by the pressure of the steam acting constantly in one direction.

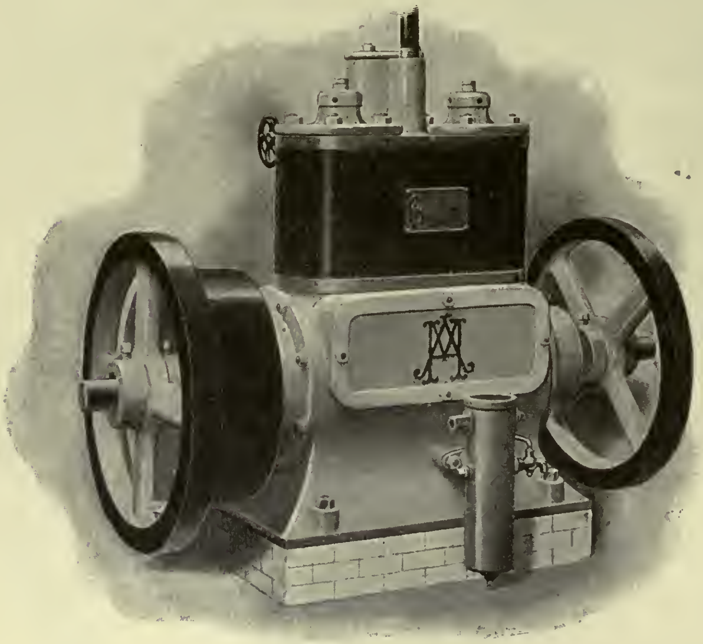


Fig. 17.—Single-Acting Engine.

The principal and most distinctive element in this type of engine is its high speed of rotation—as distinguished from high piston speed—by which it enables a much smaller engine to be adopted, smaller fly-wheels and belt pulleys instead of large massive ones with their attendant dangers and inconveniences. It also permits the engine to be used for many classes of work for which the slow-speed engine is not at all suited, such as the direct driving of dynamos and other high-speed machines. Further, the high-speed engine being so much smaller and more compact than its slow-speed neighbour, does not require such massive and costly foundations, and so saves initial cost of the plant to an appreciable extent. Another result of high speed is the reduced weight of the various parts of the engine, thus saving in the cost of transportation as well as in the lifting tackle required when erecting or overhauling on site.

As illustrating the compactness of the high-speed engine, we may mention that an engine of this type, capable of developing 100 horse-power, occupies a floor space of only 4 feet 6 inches by 9 feet 6 inches and 76 inches high, inclusive of fly-wheel and belt pulley, while the other sizes of engines are in proportion.

In Fig. 18 is shown a tandem compound single-acting engine, at one time used to some extent, and

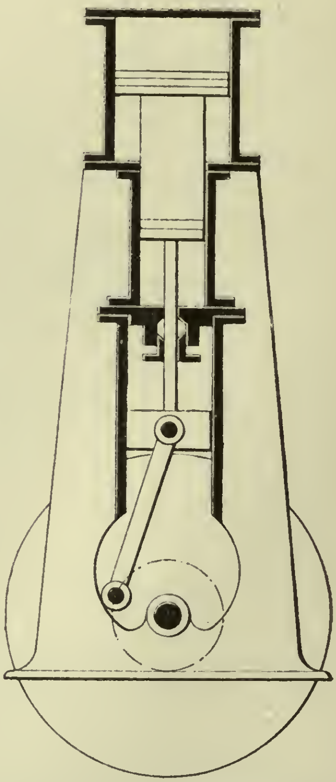


Fig. 18.—Compound Single-Acting Engine.

now again introduced with improvements. In this type of engine the steam first expands under the small piston on the up-stroke; on the down-stroke it expands from the small cylinder into the large cylinder above the piston, from which it is exhausted into the condenser or atmosphere.

The later form of this type of steam prime mover, called the Peache engine, is shown in sectional end and side elevations in Figs. 19 and 20. It has a closed crank case, and is designed for high speeds.

The engine is single acting, that is, the balance of pressure is always towards the crank shaft, so that there is no back lash in any of the bearings.

The high-pressure steam chest consists of the annular space round the main body of the valves, and is situated partly in the high-pressure cylinder casting and partly in the low-pressure cylinder casting. Steam from the throttle valve passes to a branch on the low-pressure cylinder casting, connected by passages in that casting to the three high-pressure steam chests. This construction has been adopted in order to permit of any high-pressure cylinder being removed without breaking any joint besides that between the high-pressure and low-pressure cylinders.

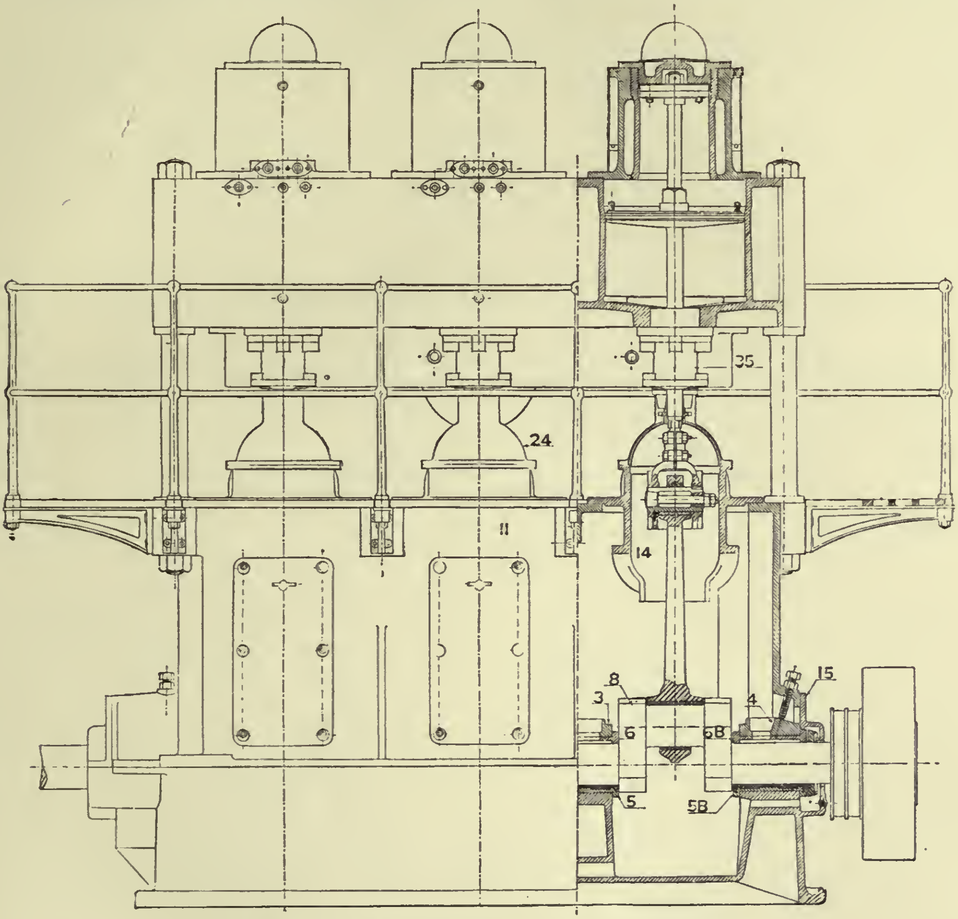


Fig. 19.—Peache Engine. Sectional Elevation.

Following the distribution of steam to the cylinders:—Steam passes under the edge of the high-pressure valve to the top of the high-pressure piston, and is exhausted over the top of the high-pressure valve, down through the centre of the main body of the valve, and thence over the top of the low-pressure valve, to the under side of the low-pressure piston, and finally it is exhausted under the lower edge of the low-pressure valve, to the exhaust chamber. In the bore of the high-pressure cylinder are cut two small bypass ports, over which the high-pressure piston passes at the top of its stroke, thus placing the space between the high-pressure and low-pressure pistons in direct connection with the upper side of the high-pressure piston. This space, which is called the controlling cylinder, is thus at the top of the stroke filled with steam at approximately the same pressure as that in the high-pressure cylinder; on the down-stroke this steam expands, and on the up-stroke it is compressed until at the top of the stroke it again reaches the pressure of the steam in the high-pressure cylinder, any small loss that may occur by leakage being made good by fresh steam passing from the high-pressure cylinder.

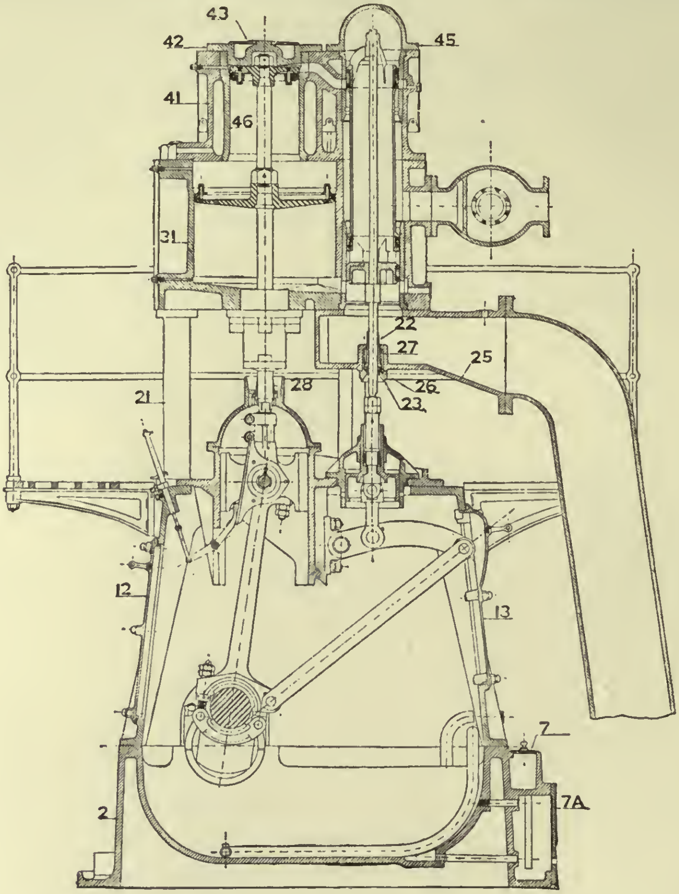


FIG. 20.—Peache Engine. Transverse Section.

The function of this controlling cylinder is to effect a balance of the upward inert force of the pistons, cross-head, connecting rod, etc., on the up-stroke; that is, to keep a slight excess of pressure in a downward direction throughout the up-stroke. Thus, on the up-stroke, the steam acts on the under side of the low-pressure piston, but instead of being allowed to act direct on the crank it does work in compressing the steam in the controlling cylinder, and in displacing the steam under receiver pressure from the high-pressure cylinder. The work thus done in compressing steam in the controlling cylinder is given out again by its expansion on the next down-stroke, and the work done in transferring steam under pressure from the high-pressure cylinder to the receiver is utilised in the low-pressure cylinder.

Referring to Fig. 21, the following is a list of moving parts of this engine:—

Connecting rods: consisting of—
51. Connecting rod and strap.
52. Big end half bushes.
53. Phosphor bronze bush.
*x*53. Phosphor bronze bush.
54. Bolts, nuts, washers, and pins.

Cross-head: consisting of—
56. Cross-head (steel casting).
57. Cross-head pin (cast iron) (not shown).
58. Bolt, collar, and nuts.

Pistons: consisting of—
61. Piston rod and nuts.
62. L.-p. piston (steel).
63. L.-p. piston rings.
64. H.-p. piston (steel).
65. H.-p. rings.
66. Distance piece. For engines up to and including G.
67. L.-p. steel cage.
68. H.-p. steel cage.
69. Piston relief springs.

Valve motion: consisting of—
71. Long link.
72. Short link.
73. Lever.
74. Joint and spring.
75A. Gudgeon pin.
75B. Pin in short link.
75C. Pin in joint.
75D. Pin in long link.
75E. Pin in long link.
76A. Phosphor bronze bush, in middle of lever.
76B. Ditto, in end of lever.
76C. Ditto, in brackets (not shown).
76D. Phosphor bronze pad, in short link.
77. Valve motion brackets (not shown).
78. Air buffer cylinder (not shown).
79. Air buffer piston (not shown).

Line of valves: consisting of—
81. Valve spindle and nuts.
82. Valve piston.
84. Lower spider.
85. Distance piece.
86. Upper spider.
87. Valve rings.
88. Valve rings pinned together.
89. Dummy ring, H.-p.
89A. Dummy ring, L.-p.

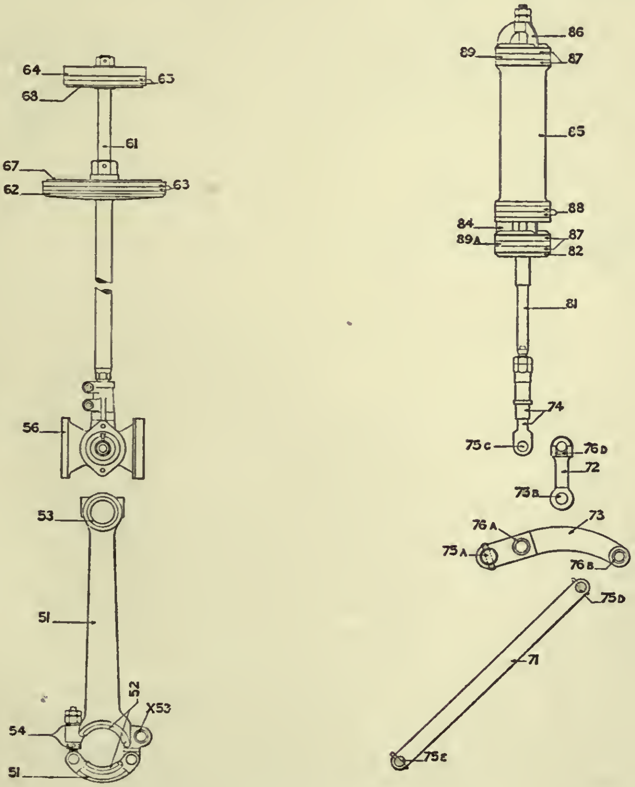


FIG. 21.—Peache Engine Details.

The governing is by throttle centrifugal governor, shown in sectional view (Fig. 22). This governor may here be taken as an example of a centrifugal throttle governor. Later on we shall deal with governors more particularly of all classes.

Referring to Fig. 22, the following is a list of the parts :—

Throttle valve : consisting of—
  91. Throttle valve case.
  92. Throttle valve cover.
  93. Throttle valve.
  94. Throttle valve spindle.
  95. Spring.
  96. Cage.
Governor : consisting of—
  101. Governor cross-head.
  102. Governor balls.
  103. Revolving spindle.
  104. Sliding spindle.
  105. Clip piece.
  106. Governor springs.
  107. Driving shackle.
  108. Links.
  109. Pins.
Governor drive : consisting of—
  112. Sleeve.
  113. Spindle.
  114. Shaft pulley.
  115. Spindle pulley.
Governor case : consisting of—
  121. Bracket.
  122. Gland.
  123. Bush.
  124. Case.
Speeding gear : consisting of—
  125. Speeding spring.
  126. Handwheel.
  127. Levers.
  128. Spring attachments.
  129. Balance weight.

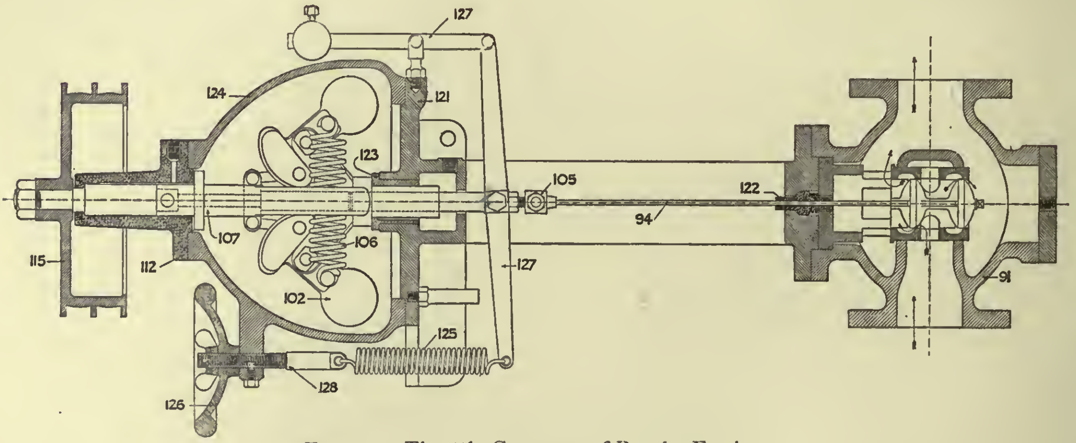


FIG. 22.—Throttle Governor of Peache Engine.

These four engines represent the commoner types now in use. All the many engines to be noticed can be traced back to these simple forms. In the first one the valve is a piston valve, with a fly-wheel governor giving a variable cut-off of the steam, whereby the speed is maintained constant at variable loads.

The Sentinel engine is similarly governed ; the others are governed by throttle, that is, by the valve of the governor reducing the steam pressure.

Two diagrams of engines working at variable loads will explain the difference between the two methods of governing—"throttling" and "cut-off."

The engine diagram, Fig. 23, shows a throttling effect. The steam is cut off from the cylinder about half-stroke whatever the lead may be, but the height to which the steam pressure may rise shown on the vertical line is governed by the load, so that only at full load can we get the full boiler pressure in the cylinder. An engine working on variable load with a throttle governor is therefore often working at perhaps half the boiler pressure, and we are uselessly maintaining, say, 100 lbs. pressure while only using 50.

Variable expansion governing is shown in diagram, Fig. 24. Here it will be seen that the cut-off line is not fixed, but varies with the load, and the steam is admitted at full boiler pressure every stroke, no matter what the load may be. By this system the steam pressure is fully utilised, and the expansion results in greater economy. A piston type of valve is generally used with this variable expansion governing, and the cut-off is varied by the fly-wheel governor moving the eccentric which drives the piston valve. But another type of valve, called a drop valve, is used for variable expansion governing, and this style of valve is worked by what is called a trip gear. The valve itself is a double-beat equilibrium valve. To admit steam the trip gear lifts it off its seats, holds it open for a period of time determined by the governor, then drops it shut. The object of this gear is to obtain a quick cut-off. All slide valves, whether flat or piston styles, close gradually. A drop valve closes instantly; hence the point of cut-off is well defined.

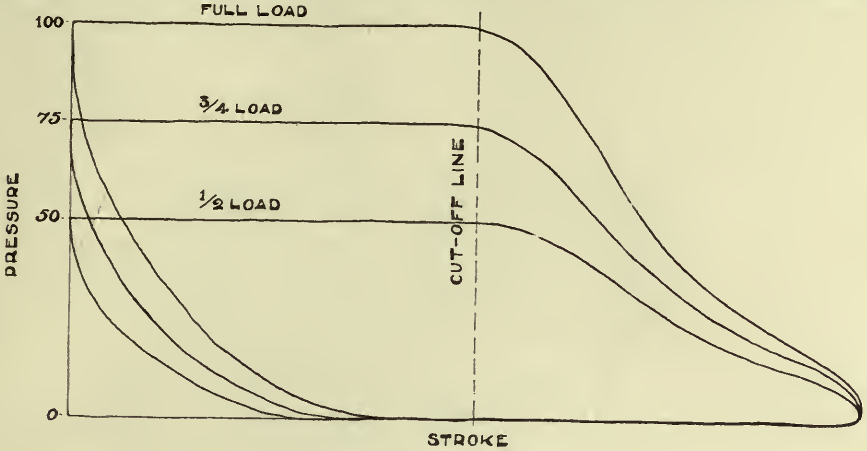


Fig. 23.—Throttle Diagram.

The piston valves are well displayed in the Sentinel and Peache engines.

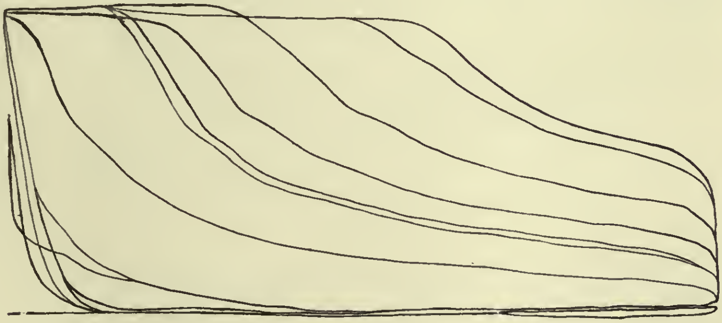


Fig. 24.—Variable Expansion Governing.

When steam engines are of large size and work with steam of high pressure the working of the slide valves becomes an appreciable part of the load upon the engine, and to obviate this various kinds of equilibrium valve gear have been introduced, the best known among them being the Corliss, with rotating valves, and the Sulzer, with Cornish double-beat valves.

Among mill and factory works the greatest diversity in engine types are to be found. In the Lancashire cotton mills and Yorkshire weaving sheds large engines of all kinds are to be found. Most of them, from a modern engineering point of view, are mere freaks—slow in speed, massively built generally, with some kind of Corliss cut-off gear. No one has any idea as to what their brake horse-power is, although a great show is made of their indicated horse-power, their vacuum, and so on. There is no standard type of prime mover among ordinary non-engineering power users. Not being engineers themselves, they are led by old-fashioned advisers and personal influences. Consequently, engines are put in with little consideration as to whether they are the best under the special circumstances of the case. But where steam prime movers are used by steam engineers for commercial purposes we find in almost every case the engine is carefully designed and constructed in all its parts, with a single eye to its purpose and the circumstances under which it is to work.

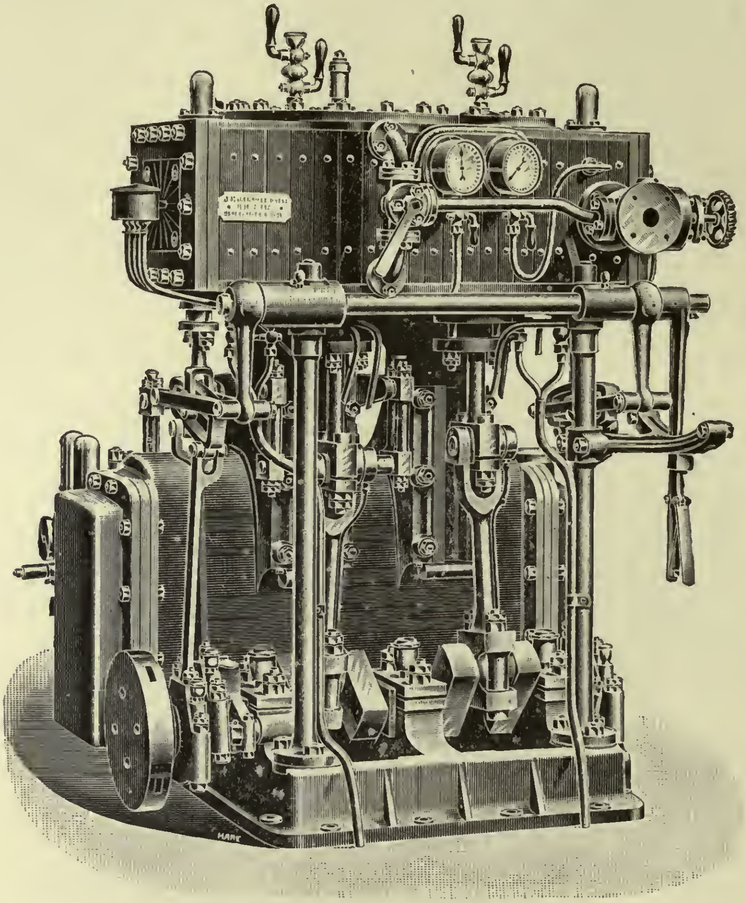


FIG. 25.—Type of Marine Engine.

This can be most obviously proved by a study of the marine engine. It is also becoming obvious in electrical power plants, although in some of these, freaks are to be found introduced by inexperienced engineers.

Perhaps the most instructive method of discussing the different types would be to class them according to the applicability to different purposes. This, to some extent, is usually the procedure, and we have thus classes such as Marine Engines, Locomotive Engines, Agricultural Engines, Traction Engines, Electric Light and Power Engines, Portable Engines, Pumping Engines, Winding Engines, Blast Furnace Engines, and the motley crowd of small and factory engines.

We may take as an example the modern small marine engine. The simplest form in use is, for reasons to be gone into presently, generally a high and low-pressure compound engine, vertical inverted cylinders, with the air pumps, feed pumps, and condenser combined. An example is shown in Fig. 25.

The engines have two cylinders, one for high and one for low-pressure steam. Pistons provided with Ramsbottom rings. Bed-plate of box form; condenser, forming part of the back standards, fitted with horizontal brass tubes. Tube plates of Muntz metal. Tubes of brass, fixed by wooden ferrules. Air, circulating, feed and bilge pumps, with important details of gun-metal. Link motion reversing gear with cast-iron eccentric pulleys, cast-iron straps, links, and valve spindles of steel. Back standards cast iron; front columns of wrought iron. A special mixture of sound cast iron is used for castings. Bushes throughout the engine are of best brass. Crank shaft, connecting, and piston rods are of best mild steel. A complete set of lubricators is provided.

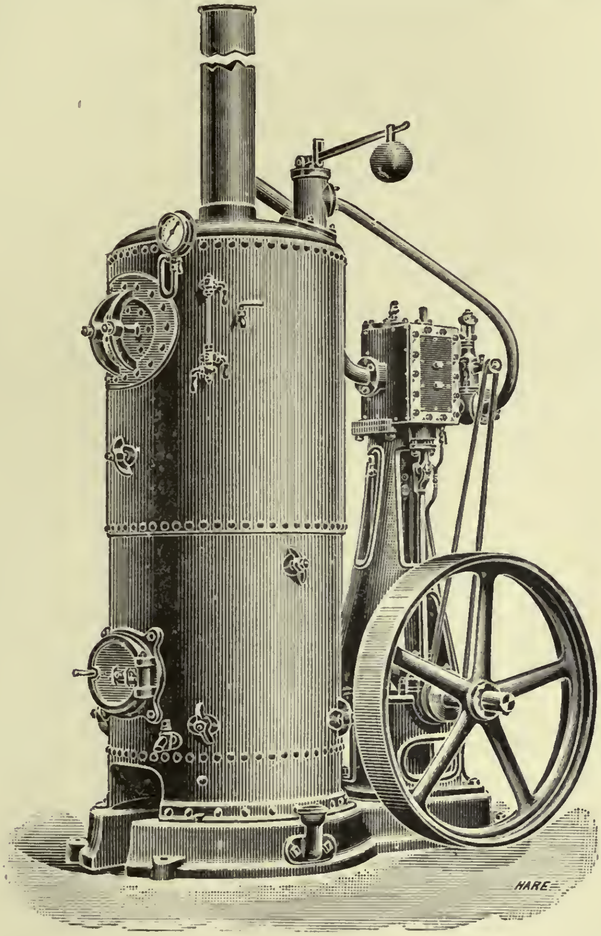


Fig. 26.—Combined Vertical Engine and Boiler.

The vertical engine, with upright boiler, continues to be largely used both at home and abroad for agricultural and miscellaneous purposes. The combination economises space, and saves expense for foundations or building in. The design here presented is entirely new, and the machinery is in every respect equal to modern needs.

The engine and boiler are proportioned to work with a steam pressure up to 75 lbs. per square inch, and the extra capacity of the latter will admit of miscellaneous fuel being used. The pump is arranged to draw feed water from the hollow bed-plate.

The working parts, which are few and simple, have large wearing surfaces, means of adjustment, and provision for lubrication. The crank shaft admits of power being taken from either side. The fly-wheel is turned to receive a belt.

The boilers are made of mild steel, and have full set of fittings.

Fig. 26 represents this common combination.

In this introductory chapter we have briefly considered the outstanding features of the elementary types of engines. It will be necessary in next chapter to examine the elements of the various types of more elaborate engines before describing the various engines under the classification already given.

CHAPTER II

STEAM ENGINE PARTS AND ACCESSORIES

We have referred to a few of the various parts of a steam engine already. A closer study of them will now be made.

THE CYLINDER AND PISTON

This has remained much the same in all engines since James Watt's time. Cast iron is the best material for steam cylinders. They are usually lined with a thin cylinder of harder material, generally cold-blast cast iron. This takes on a polish, and is not readily scored. This liner also makes the steam jacket a simple matter, by leaving an annular space between the jacket and cylinder wall. The construction of the cylinder and liner is seen in Fig. 27, cylinder section. At each end of the cylinder is a fitting strip about ½ inch smaller in bore than the cylinder, and on each end of the liner is another fitting strip ½ inch larger in diameter than the liner, and these strips are bored and turned to an accurate fit, so that a 1-inch open space is left between the liner and the cylinder, into which live steam is introduced to keep the interior of the cylinder hot. This space is therefore the hot steam jacket. There are various methods for fixing the liner steam-tight in the cylinder. One method is shown in Fig. 27, a section of a marine engine cylinder and piston valve. The liner has an internal flange at the bottom, which bolts to the inner face of the cylinder end. One of the bolts can be seen in place.

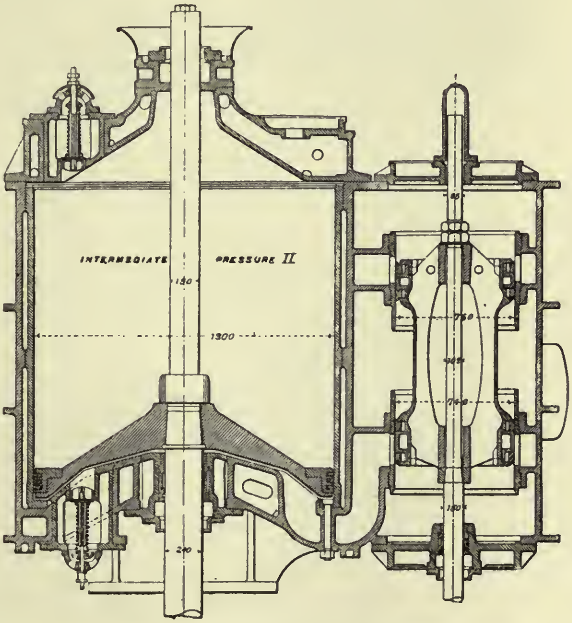


Fig. 27.—Steam Engine Cylinder.

Both cylinders' ends may be bolted on, but generally one end, that which the piston rod works through, is cast in one with the cylinder barrel. The other end cover is made hollow in large sizes, with stiffening ribs, as may be seen in Fig. 27. The covering end is bolted on or fixed by studs. In some cases with large pistons the piston rod is prolonged through the upper or back cover to steady the heavy piston, as shown in Fig. 27. In the end covers relief valves are fitted, consisting of valves loaded by springs, to allow of the escape of water when it collects in the cylinder. When the piston arrives at the end of the stroke it forces out any water over that which would fill the clearance spaces.

The piston is always fitted with packing rings, or spring rings, as they are called, in order to make it steam-tight. In the old atmospheric engines the packing was of old junk (old tarry rope), and hence the term for a packing ring in a piston. "The junk ring" (Fig. 28) is an interesting photo of an old piston from a Newcomen steam engine of beam type. The junk is seen wrapped around the piston, and held down by heavy iron weights with handles cast into them.

Fig. 29 shows a modern form of spring packing—the Lancaster. It consists of a spiral spring laid in between two split rings. In some cases of small cylinders the split rings have sufficient elasticity in themselves to act as packing.

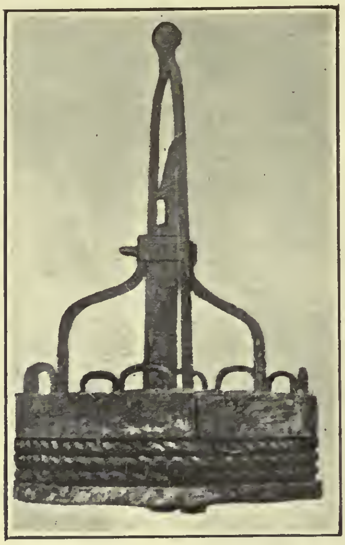


Fig. 28.—Junk Packing

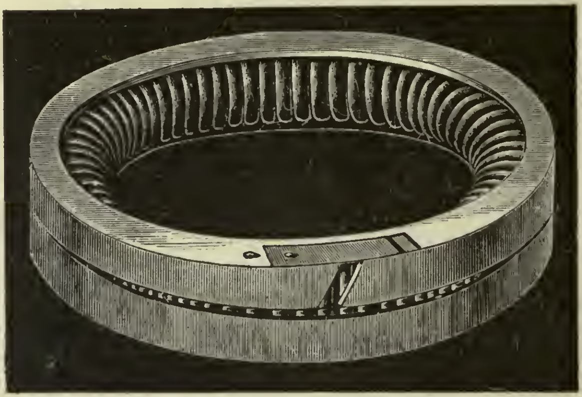


Fig. 29.—Spring Ring Packing.

The illustrations (Figs. 30 and 31) show a device with the object of minimising the friction of piston rings on the walls of the cylinder, and also of confining the possible

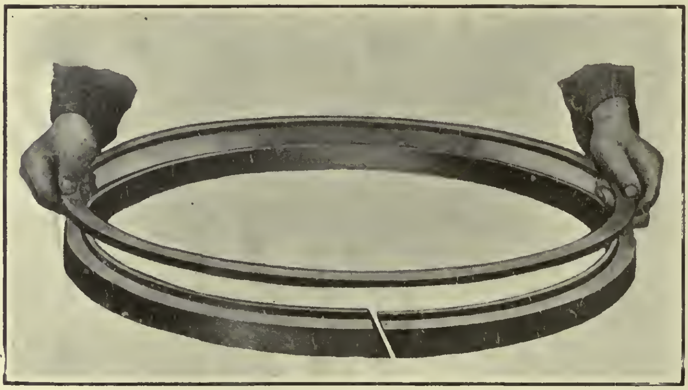


Fig. 30.—Spring Ring with Limit Ring.

wear of ring and cylinder within certain predetermined limits. The limit ring consists of a flat uncut ring, which fits over a shoulder left on the face of the piston ring. When the latter is placed within the cylinder the inner diameter of the limit ring is slightly

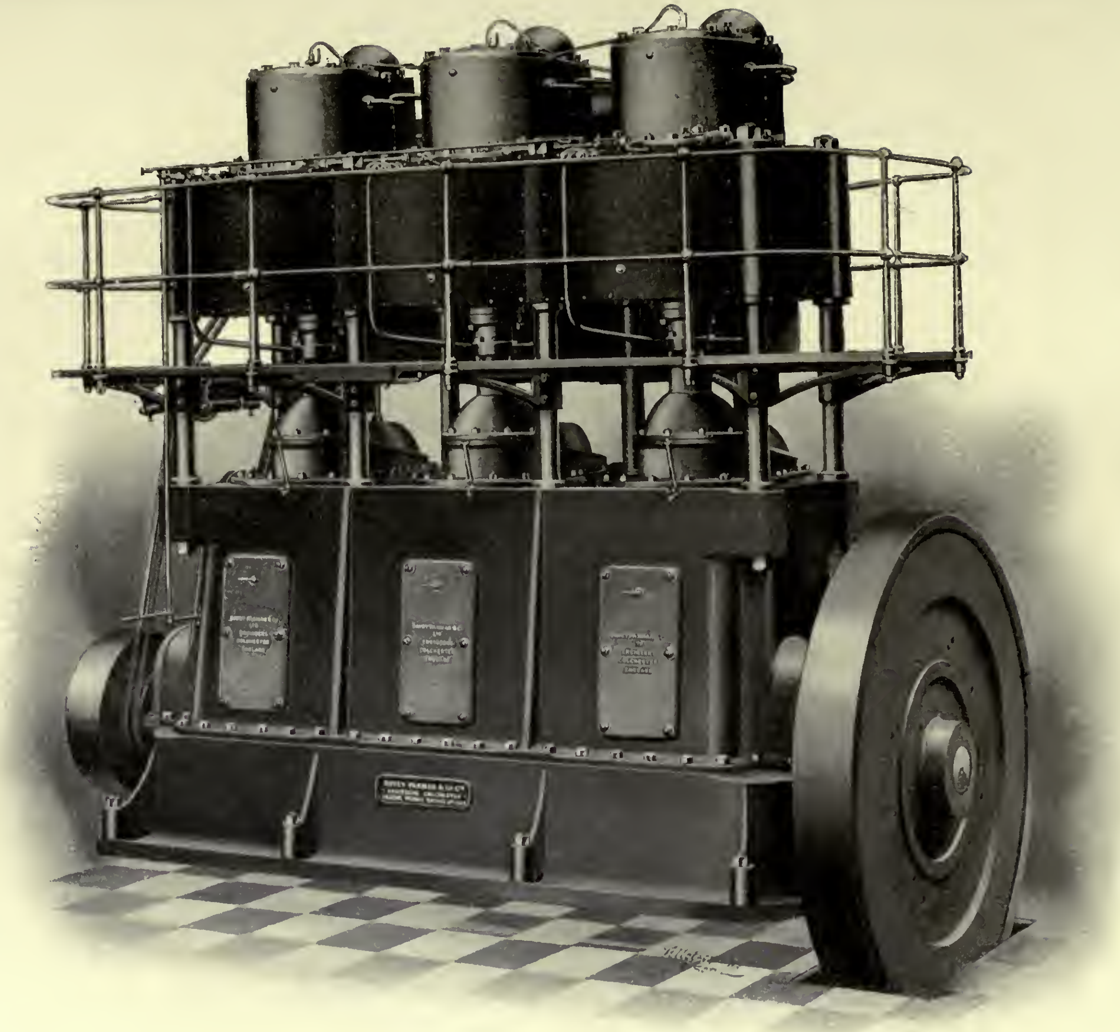


Plate I.—Three-Cylinder Peache Engine
(Davey, Paxman, & Co.).

greater than the diameter of the shoulder over which it fits, and the wear of the ring and cylinder can only continue until these diameters become equal. As the limit ring tends to cause the piston ring to maintain its cylindrical form, it is found that the accuracy with which the piston ring fits the cylinder when the limit of wear has been reached is such that both the steam efficiency and the mechanical efficiency of the engine are increased.

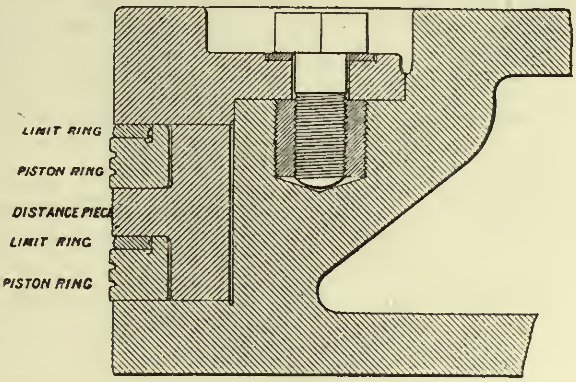


Fig. 31.—Limit Ring.

It will be noted as a further advantage that the limit ring acts both as a piston clip when inserting the piston, and does away with the necessity for the troublesome joint plate which has to be fitted to rings of the coiled-spring type; and further, there is no possibility of pieces of a broken ring getting in the ports when withdrawing the piston for examination.

The inner surface of the limit ring is grooved so as to reduce the amount to be filed or scraped for purposes of adjustment. The device is, of course, applicable to all pistons of the junk-ring type, or those made in halves, and is also supplied for piston valves and high-pressure pumps.

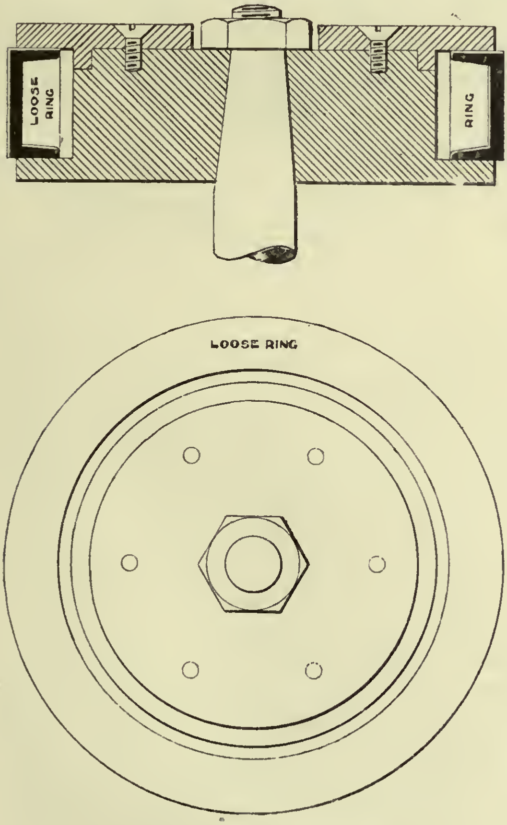


Fig. 32.—Unsplit Ring.

Messrs. Richardsons, Westgarth, & Co. Ltd., Hartlepool, are the makers, and it is known as Morison's limit packing ring. There are a great many patented piston rings, especially for large pistons. Experience alone can decide which are best.

Some time ago we came across engines without any spring rings or spring packing in their pistons, vertical inverted cylinder engines, and they were perfectly steam-tight. The cylinders were accurately bored, and the packing ring accurately turned to fit. Being somewhat broad, the ring fitted into the piston like a junk ring (Fig. 32). They had been working for over a year, but whether they kept tight for longer periods has not been ascertained. This unsplit ring would not succeed in an engine in which the cylinder wears oval, owing to side pressure of the piston, but for vertical cylinders with easy fitting pistons there seems no reason why it should not make a good fit for a long time.

The packing of pistons has formed a great field for improvers, and many devices are to be found in practice; but the spring ring in some shape or form is the basis of all of them, with the above exception.

The use of higher steam pressures and superheating has been greatly extended by perfection in metallic packings, both for piston and valve rods.

The temperatures employed in steam cylinders are much lower than those used in internal combustion engines; but still, metallic packings have been designed to withstand even the high temperatures of gas engines.

The ingenious packings devised from time to time by Mr. C. A. Parsons for his steam turbines are well worth studying. The simple "corrugated" system of packing, that is, by a series of rings, has been found effective for both dummy pistons and high velocity shafts.

An effective spindle packing applied to centrifugal pumps is illustrated in Fig. 33, made by Messrs. Belliss & Morcom of Birmingham. It would serve also for low pressure or vacuum ends of steam turbines.

With ordinary packing, as soon as an attempt is made to raise water from any depth, the packing has to be continually nipped up so as to preserve the vacuum in the pump casing. As a consequence, the shaft is rapidly worn down, and the pump has to be continually attended to to prevent its failure. In the fitting illustrated the gland is done away with, and the air-tight joint is secured by means of a bearing against a collar on the shaft, as shown in Fig. 32, in which A is the collar, and B is the air-tight joint; C is a flexible disc, and D an indiarubber joint ring. Inside the flexible disc there is a vacuum space E, connected by grooves with the suction side of the pump, one of the grooves being shown at F. The cock G is connected to the discharge branch of the pump, and clear water supplied to cool the white-metal bearing, and prevent the dirty suction water getting back to score the bearing. This cock can be adjusted so as to balance the pressure due to the vacuum against the white-metal face. The lubricator shown supplies lubricant to the collar. The operation of letting the water in behind the back of the bearing not only prevents the excessive wear of the surface of the air-tight joint and disc, but keeps the bearing cool and clean. It has been found of special value when sandy or dirty water was being pumped, as the sand or dirt would not then go to the bearing. It should be noted also that, with the same overall length of pump, a good solid bearing is secured in place of the packing gland and small bearing.

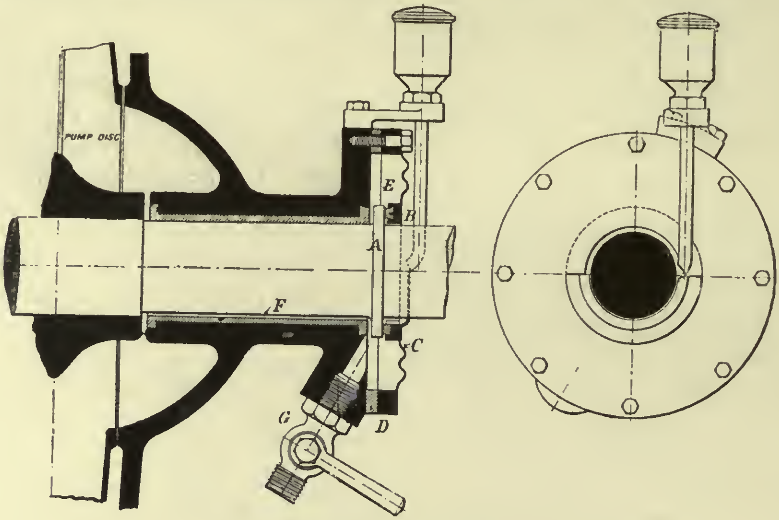


FIG. 33.—Suction Packing.

Messrs. Belliss & Morcom say that they have a number of pipes already fitted with this device at the Poplar Central Station. They are working from a depth of

12 feet to 18 feet, and by means of the arrangement illustrated a vacuum of 28 inches has been maintained in the pump casings. It was found impossible to secure such a vacuum with the ordinary type of soft packing.

The Standard Piston Ring and Engineering Company, of Sheffield, make rings specially for oil, gas, and petrol engines. Small rings for these engines are made perfectly true and cheaply. Many rings for petrol engines are only $\frac{1}{4}$ inch deep, $\frac{5}{32}$ at the back, $\frac{3}{32}$ at the cut.

In order to get equality of pressure against the cylinder all round, these rings are turned in batches, about a dozen each being held in the jig (Figs. 34 and 35), in such a way that they are turned so as to be a perfect circle on their periphery when sprung into the cylinder. The rings are first placed in a cylinder, which exactly represents the engine cylinder in which they are to be used, having to be compressed to fit this cylinder, so that they will spring out sufficiently to make a steam or gas-tight joint. The object of the jig is to hold them in their compressed position while they are being turned. There is a base-plate, grooved as shown.

A plug template is used, round which the rings are placed. The base plate is perforated at the end, and through it there passes a bolt; at the other end of the bolt there is a second plate, slightly smaller in diameter than the outside diameter of the rings as compressed. When the rings are in position in the cylinder they are held firmly between the base plate and the other plate mentioned, which also fits over the bolt. By setting up a nut the rings are pressed together endwise, and this holds them in the same position which they occupy in the cylinder when the latter is withdrawn. The rings can now be turned up, and naturally, when they are placed in the engine cylinder, they will be perfectly circular. A large number of rings of this character have now been supplied for motor car and gas engine purposes, and have given very satisfactory results.

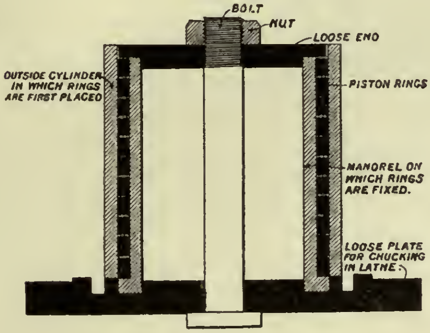

BOLT
NUT
LOOSE END
OUTSIDE CYLINDER IN WHICH RINGS ARE FIRST PLACED
PISTON RINGS
MANDREL ON WHICH RINGS ARE FIXED.
LOOSE PLATE FOR CHUCKING IN LATHE

FIG. 34.—Spring Ring Jig.

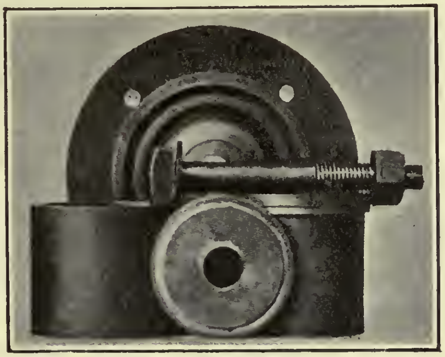


FIG. 35.—Spring Ring Jig.

They also make a spiral split ring, as illustrated in Fig. 36, suitable for large pistons.

The blanks from which these rings are made are solid ring castings. The first operations are to turn and bore them cylindrically; after this they are sawn almost completely round, leaving only a short piece, which is known as the "bridge," and this is shown in the illustration. The joints are then milled out to form the overlapping parts; after this the rings are put into a template and faced up so as to make the depth equal all round, and exactly the width of the groove in the piston. This operation corrects the unequal depth of ring which would be due to the metal removed by the saw. The two parts of the ring, where it is split, tend to come apart the thickness of the saw cut, and this causes the ring to fit tightly in the groove in the piston. Taking off the superfluous metal in this way also makes the rim tapering; that is to say, it is somewhat thicker at the part where it springs from the bridge than it is at the end. These rings can be fitted into a cylinder very easily, and will yet keep tight, and the half-lap joints are also practically steam-tight. The split rings are turned up in a jig similar to that described for the solid rings; they are thus made to the form they will assume in the cylinder, and will be a fit all round without undue pressure at any one part. The split piston ring has been made for cylinders up to 6 feet in diameter, and over 200 lbs. to the square inch pressure. It is possible to spring them over solid pistons from 4 inches in diameter up to 6 feet, and they may be made for use with any existing piston. These rings are made both for steam engines and internal combustion engines; for pumping work and other special operations they are made of brass. They are found to wear the engine cylinder very evenly all round, so that it remains cylindrical.

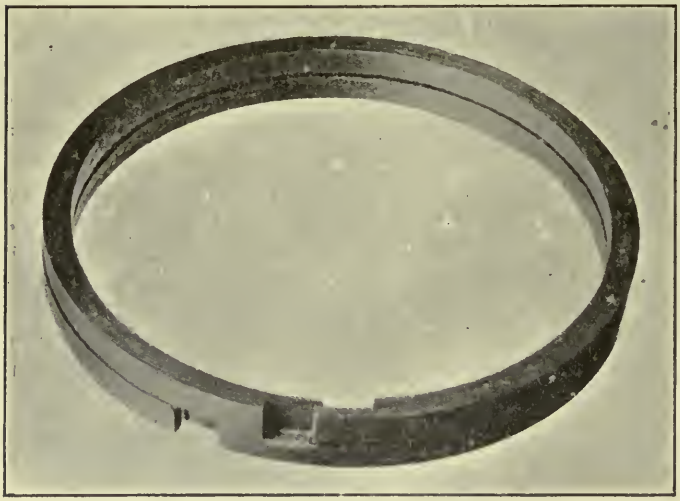


FIG. 36.—Spiral Spring Ring.

The well known Ramsbottom rings are made also by this firm in a very simple and satisfactory manner. They are a cheap ring, but only preferable to the cast-iron rings in cases where sudden jars and blows are to be met with, such as steam hammers, hydraulic machines and pumps, wherein cast-iron rings would break.

The steel rings are formed of cold-drawn steel rod, which is made exact to gauge, to within $\frac{3}{1000}$ inch, under or over, that being the allowance made. The rod is bent up in rolls to a coil, which is afterwards sheared so as to produce the separate rings. These are flattened by hand on a plate. Over 400 different sections are made for various purposes, ranging from $\frac{1}{8}$ inch square up to $1\frac{1}{2}$ inch by $\frac{3}{4}$ inch.

The piston rod is most commonly fixed in the piston by a tapering end, and a nut with a lock nut, or by a nut and a key or pin. It is important that this fastening should be secure, and yet easily removed to get the piston off easily. A loose piston is not uncommon, although a serious defect. The piston rod passes through a stuffing box in one end of the cylinder, and sometimes is prolonged to pass through both ends in stuffing boxes.

For small powers and low pressures a common stuffing box is used with success, stuffed with rope yarn, asbestos, or other fibrous material. For high pressures, and in nearly all steam engines nowadays, some form of metallic packing or stuffing is used. The packing shown in Fig. 37 is a metallic packing by Messrs. Murray & M'Vinnie, Glasgow. Many forms of metallic packing are in use; all of them are based on the spring ring principle so successfully applied to pistons.

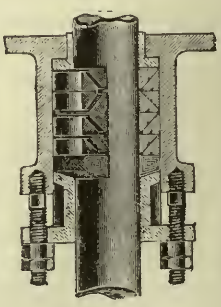


FIG. 37.—Metallic Packing.

They are more costly, and not so cheaply replaced as fibrous packing, but fibrous packing is only applicable at pressures under 100 lbs. per square inch; above that pressure, and with superheated steam especially, metallic packing is superior, and worth the extra cost.

Dr. Proell's metallic packing is shown in Figs. 38 and 39. This packing is simple in construction. It consists of a series of alternately solid and split cast-iron rings, which are interchangeable, and which fit closely into one another and round the piston rod. The packing is so constructed that it will yield to any lateral motion of the rod, and it does not depend for its tightness on ball joints, but on a series of movable flat joints ; and one of the advantages claimed for it is that tightness is secured without any excessive pressure or friction. The method of arranging the packing rings will be clearly seen by reference to Figs. 38 and 39. In these views the solid cast-iron dished rings are shown at A. The outer diameter of these rings is made to fit accurately into the packing chamber of the cylinder cover, while the inner diameter is slightly larger than that of the rod to be packed. The rings A are finished and accurately ground by special machinery, so that the annular spaces C are of exactly the same size. The solid rings A are fitted into the packing chamber, and one split ring B is placed in each of the compartments C. The split rings are bored inside to the exact diameter of the piston rod, and their thickness is exactly that of the depth of the annular space C. In outside diameter they are slightly smaller than the solid rings A, so as to allow of some lateral motion, the split rings being free to move sideways in every direction to allow for any play there may be in the rod. The three segments of the split ring B are held together by a thin spiral spring (Fig. 39). This view also shows the arrangement of the three segments of the split ring. The ends of the spiral spring are hooked together so as to form a ring. When the required number of solid and split rings have been placed in the packing chamber the whole of them are firmly

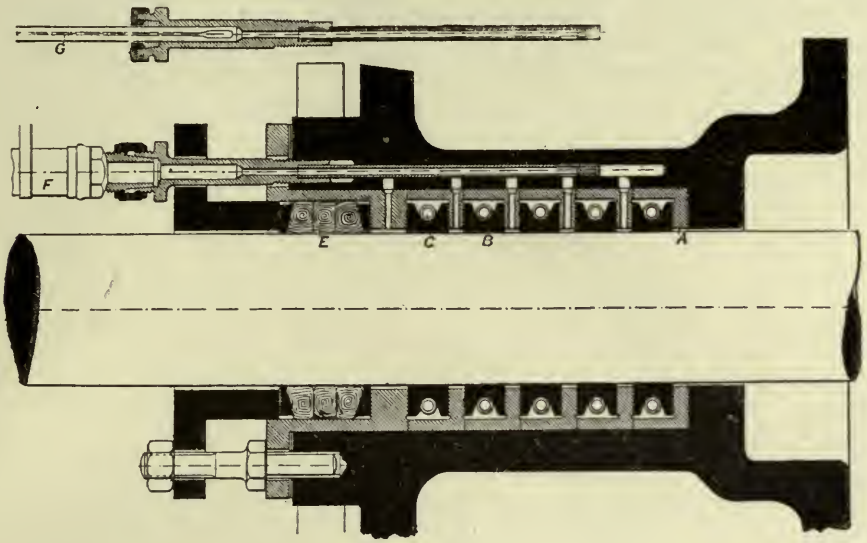


FIG. 38.—Proell's Metallic Packing.

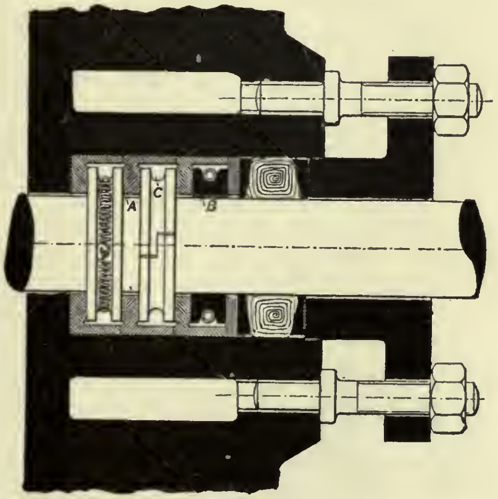


FIG. 39.—Metallic Packing.

pressed together by a gland E (Fig. 38). It is not claimed for this packing that one joint is of itself steam-tight, but that by building up a series of rings a number of minute passages are formed, in traversing which the steam gradually falls in pressure and temperature.

This has been proved by a very interesting series of tests, carried out by means of the indicator and thermometer attachments shown in Fig. 38, F being the attachment for the indicator, and G the one for the thermometer. By means of the indicator a series of diagrams have been taken from the different rings of one of these metallic stuffing boxes, the results of which are shown in Fig. 40. The engine experimented upon was one of 700 horse-power, running at 100 revolutions per minute, and fitted with a Schmidt superheater. The tests were taken from the high-pressure cylinder. The temperature at each ring is clearly marked opposite each diagram. These temperatures and pressures form a most interesting record of how the temperature and pressures falls from the inner ring of the packing to the one farthest from the steam. It will be seen, on reference to Fig. 40, that the steam pressure fell from 135 lbs. per square inch at the inner ring to 0 at the outer, while the temperature fell from 620° Fahr. at the inner ring to 300° Fahr. at the outer one, showing how rapid the drop was after actual contact with the superheated steam had ceased.

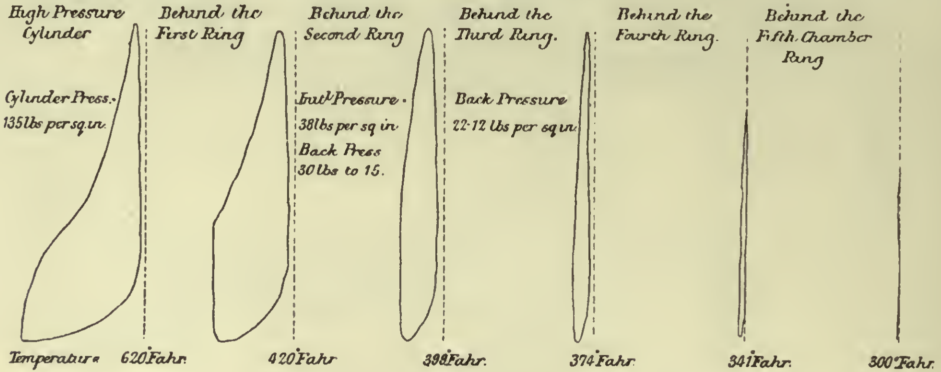


FIG. 40.—Pressure Diagrams of Proell's Metallic Packing.

An excellent design of ring packing invented by Mr. S. A. Ward is shown in Fig. 41. The packing consists of three rings, two of which are split and the other solid; and all three are kept in place in the piston by a junk ring, as shown in illustration. Fig. 41 is an elevation and section of the piston, and shows the position of the bolts which secure the junk ring.

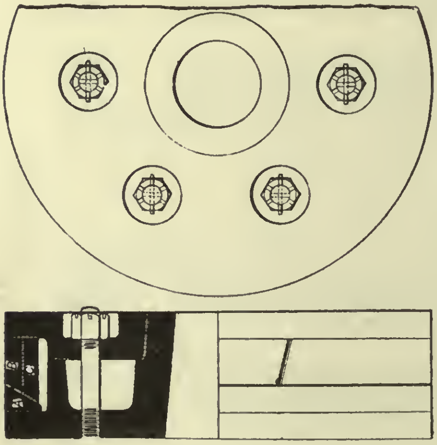


FIG. 41.—Spring Rings.

The ring A, shown in section in Fig. 41, is turned an easy fit to the cylinder, and has one side bevelled. The ring B is turned the exact size of the cylinder, and is bevelled at the same angle as A. It is then sawn through. The ring C, which constitutes a strong spring, is turned to a slightly larger diameter than the inside of B, and is sawn through and fitted inside the latter as shown. The saw cuts in B and C are not placed opposite one another in the piston, and all three rings are held in place by the junk ring, as before stated.

The advantages claimed for this arrangement are the following. Friction between the ring B and the cylinder is reduced, because steam getting within the block cannot blow the ring B out to against the cylinder, the ring A being solid, and strong enough to resist the pressure of the steam which is transmitted through the bevel on A, and is taken by the sides of the piston block; so that the tendency is to keep the rings a perfect fit sideways in the block. The little wear that always takes place in the block allows the rings to expand in diameter, and so take up any slight wear there may be in the diameter of the cylinder. On account of the full force of the spring ring C being resisted by the ring A, the wear of the cylinder by the rings is consequently almost entirely prevented. The frictional contact between the two broad rings prevents their being easily crushed in by the pressure of steam, yet they are quite free to move in the piston head, and allow for any little irregularity in the alignment of the engine. Another advantage claimed is, that there are no steel springs and small parts to break and cause accidents.

These pistons have been found to work remarkably well under the high temperature of superheated steam, and the friction, as compared with that of an ordinary piston, is very much reduced; in fact, a 20-inch piston fitted with these rings can be pushed backwards and forwards in the cylinder with one hand, yet when coupled up and full boiler pressure turned on the rings are perfectly steam-tight.

The rings are made of soft cast iron, and are equally suitable for high and low-pressure pistons and piston valves.

The plain spring ring is perhaps more commonly used than any other; these rings are naturally in order to give an equal spring made thicker at the back than at the cut opening. They are turned eccentric, the inside to the outside, and when cut and sprung into the cylinder are not truly circular. Many devices called jigs have been devised for truly turning up the exterior of the ring to a true circle to fit the cylinder after they are cut.

Messrs. Lancaster & Tonge make another form of metallic packing shown in the sectional views (Fig. 42); it is double box type. The upper set, it will be seen, fits inside the stuffing box, and consists of alternate brass and white-metal rings R, fitted in a cast-iron casting S. The rings R are pressed into the casings by a number of springs in the bush Y_1, causing them to bed to the piston rod and the inside of the casing. These springs also keep the casing firmly bedded against the dividing plate T, which separates the upper from the lower set of packings. The latter are contained in a casing or box C, which is secured to the front of the stuffing box by studs, as shown, making a steam-tight joint between the two sets. The two rings U and V, which rest on the bottom of the casing C, form a ball joint, which allows for lateral or other motion of the rod. On these rest the packing rings W and W_1, which are each divided into four segments, as shown in Fig. 42, and are pressed to the rod by the encircling springs Z. The latter are furnished with a hook and eye at each end, and are made in halves

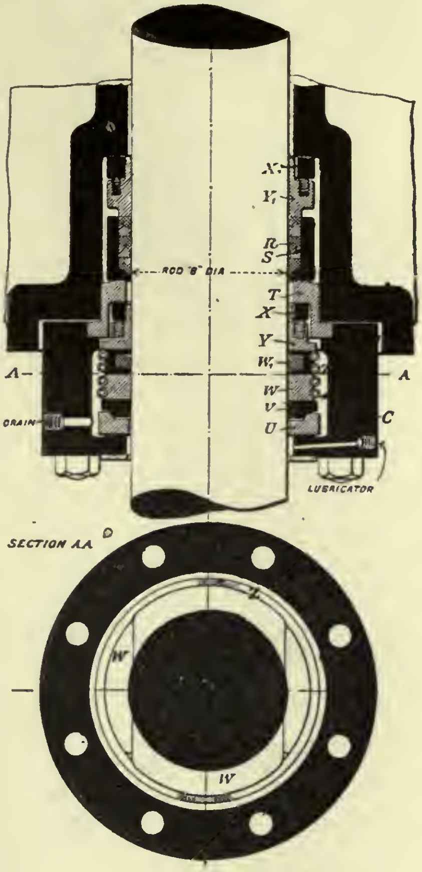


FIG. 42.—Metallic Packing.

screwing into each other, so that the length of the springs can be reduced as the packing blocks W wear. The lateral joints between the rings U, V, and W are kept steam-tight by a number of springs in the bush Y, which react in the washer X, and press them down on to the bottom of the casing C.

To the outer end of the piston rod is fitted some form of a joint upon which the connecting rod swings; and also guide blocks or planes or cylinders to take the side pressure due to the angularity of the connecting rod. In old-fashioned engines the guide blocks, sliding between guide rods, were carried on a cross bar fashioned on the end of the piston rod. This bar was called the "cross-head," and upon it also was attached the connecting rod ends. It is far better to support the piston rod end in a block or blocks immediately beneath it, but however it is designed, the old term cross-head is worth retaining. A good example of a cross-head is shown in the drawing (Fig. 43) of Messrs. Robey's engines. Cross-heads are generally made of mild steel castings; sometimes, however, they are forged solid, and then milled and bored out to size and shape. And as they are subjected to the whole stress and strain, it is important that they should present a large factor of safety. The surfaces which rub on the slides are of cast iron, as that metal wears better under friction than any other; these surface plates or slippers, as they are sometimes called, are fixed by screws and readily replaced. The pin upon which the connecting rod oscillates, sometimes called the "gudgeon pin" and sometimes the "wrist pin," requires considerable care in designing the manner of its fixing in the cross-head. It must be a good firm fit, no shake, and yet be easily removed.

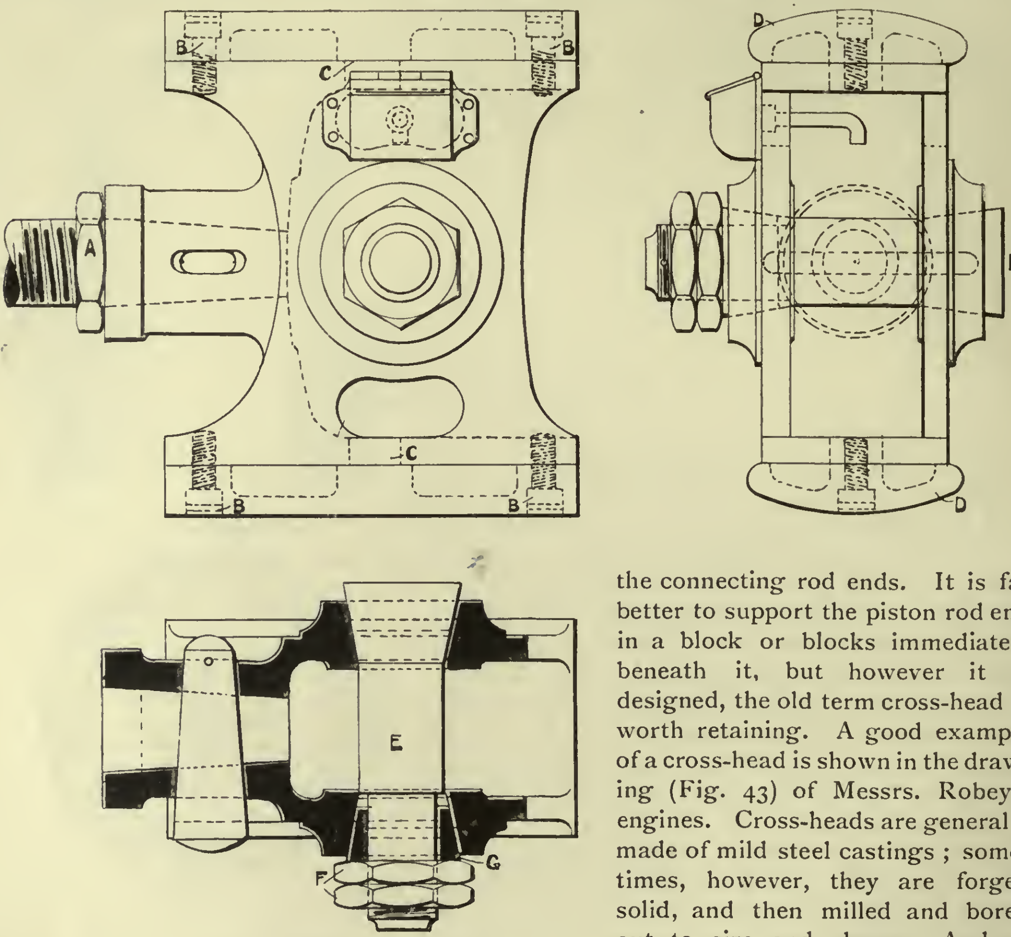


FIG. 43.—Robey Engine Cross-Head.

In some cases the cross-head is guided by flat-planed slides, in others by bored cylindrical slides. The one is as good as the other, so far as guiding is concerned. It is a question for the manufacturer. Generally speaking, it is cheaper to adopt bored slides, when they can be truly bored concentric with the cylinder. The slippers of the cross-head in bored slides consist of curved plates screwed on and turned up truly to fit the bore. These slippers have a very large surface, so that the wear is exceedingly small; but when they do become slack they can be packed up with thin sheet steel.

In order to secure the wrist pin the drawing shows clearly the design for this purpose. The pin has a taper head on one end and a screw and taper steel collar fits over the other end, the cross-head being bored out to fit the two tapers. The collar is split lengthwise, so that it can be forced in a close fit on the pin by a screw and two nuts.

The piston rod is fitted into a tapered bore in the cross-head, and secured by a collar as shown in Fig. 43.

This type of cross-head is designed for single-ended connecting rods, the connecting rod end being fitted into the jaw of the cross-head. Preference, however, is often given to the cross-head shown in Fig. 44 solid, and to which a double-ended connecting rod, as shown in Fig. 45, can be fitted. In this connecting rod end the brasses can be taken up in the usual way, and the pin is fixed by cottar in the cross-head. This is the commonest type of cross-head and connecting rod for all large powers of double-acting engines.

The strains upon high-speed connecting rods are principally three in number,—compression, tensile, and bending; and to resist the latter it is found best to make the rod tapered larger at the crank end. In old-fashioned slow-going engines the rod was gracefully tapered from both ends, being thickest in the middle, the rule being—

Diameter in middle, $0.022 \sqrt{104 \sqrt{P+P}}$.
$L =$ Length of stroke in feet.
$P =$ Pressure × area of piston.

This holds good up to 100 revolutions per minute, but for 150 the constant should be 0.025 beyond that speed, the rod should taper right up to the crank end. The maximum strain on the connecting rod end should not exceed 4500 lbs. per square inch section.

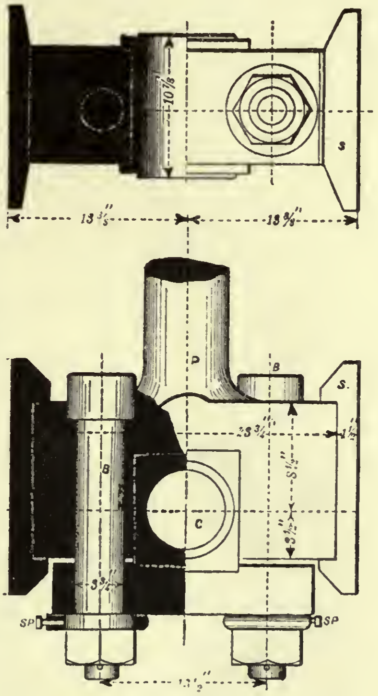


Fig. 44.—Solid Cross-Head.

In single-acting engines the connecting rod is only subject to compression and bending strains. The Westinghouse engine connecting rods (Figs. 46 and 47) are fair examples these.

In the Standard engine two forms of connecting rod are used. In one form, in the smaller sizes, the body and cap are drop forged from steel, and are connected by a hinge joint. In both forms the upper end is bored out and fitted with a renewable babbitt metal bushing, in which suitable oil grooves are cut. It is not necessary to make any adjustment at this joint. All strain on the bearings being in one direction, downward, the slight amount of lost motion due to wear has no effect whatever on the operation of the engine, and gives no sign of its presence. When the bushing is worn out a new one can be obtained at a small cost, and the operation of putting it in place is very simple. The lower end of the rod is babbitted to fit the crank, and, if desired, wear can be taken up by removing one or more washers from the bolt which keeps the hinged cap closed over the crank pin. When the babbitt is entirely worn out the rod may be rebabbitted by the owner using a suitable jig, or it may be returned for rebabbitting. At any rate, it is a problem which does not ordinarily come up oftener than once every five or six years.

On the larger sizes (Fig. 46) a hammer forged steel body is used, to the lower end of which is fitted a horseshoe-shaped strap. This strap is not adjustable, but is held rigidly in place by two turned tap bolts passing entirely through the stub end of the body, and threaded into one side of the strap. These bolts are easily accessible for removal through the opening in front of the crank case. In the bottom of the strap is fitted a semicircular bronze liner, and above this is a brass box lined with anti-friction metal.

In single-acting engines the rod is kept in constant compression by various devices employing compressed air or steam cushions to take up the momentum of the upward flying piston and rod.

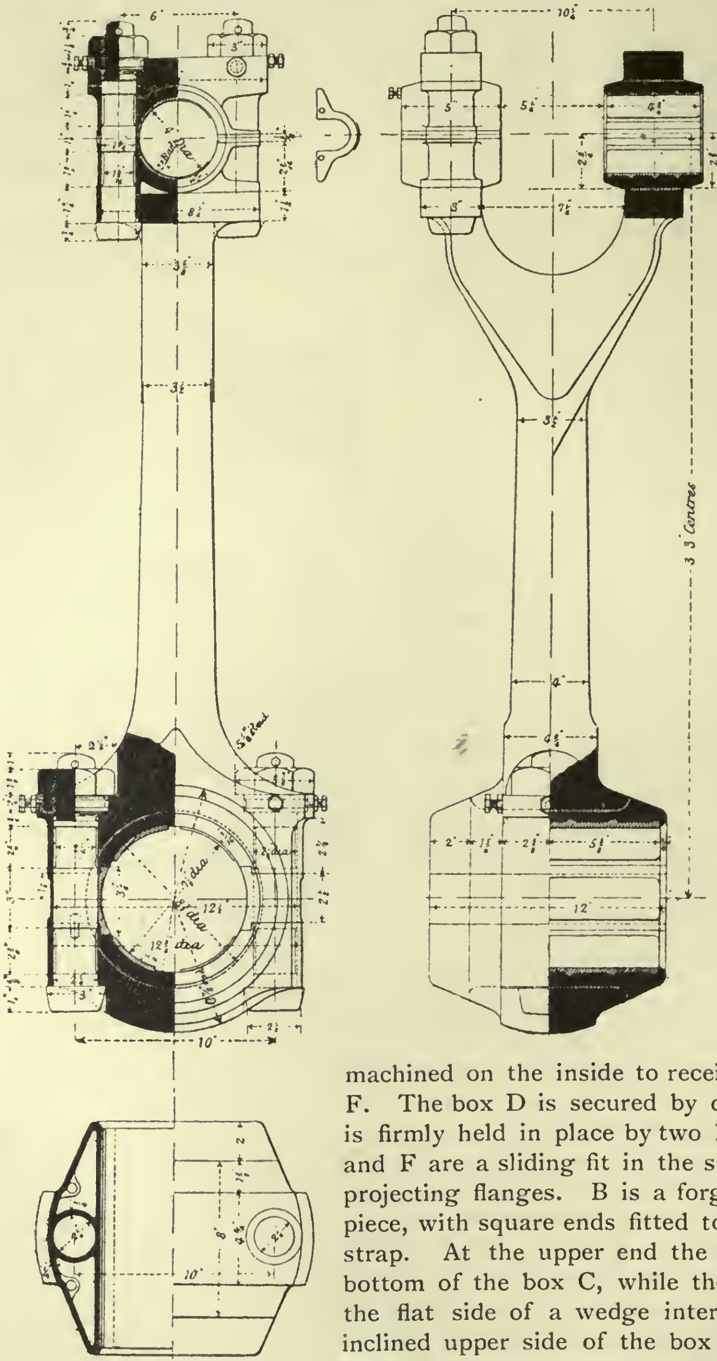


FIG. 45.—Connecting Rod.

The other form of Westinghouse connecting rod, shown in detail in Fig. 47, consists of a continuous forged strap A, extending the entire length of each side and around the top, and machined on the inside to receive the boxes C, D, E, and F. The box D is secured by dowel pins, and the box E is firmly held in place by two heavy bolts. The boxes C and F are a sliding fit in the strap, and held sidewise by projecting flanges. B is a forged steel strut or distance piece, with square ends fitted to the inner surfaces of the strap. At the upper end the strut B bears against the bottom of the box C, while the lower end bears against the flat side of a wedge interposed between it and the inclined upper side of the box F. Owing to the single-acting principle involved the boxes D and E merely act as keepers, all wear coming on C and F. By drawing up the wedge with its bolt, readily accessible through the opening in the crank case, the box F is forced downward, and through the strut B the box C is forced upward, thus taking up the lost motion at both ends simultaneously and equally with a single adjusting screw. When the boxes are worn sufficiently to allow the wedge to go to its extreme travel it is set back again, and one or more sheet steel plates or "shims" placed between it and the end of the strut to compensate for the wear.

Boxes C and D are solid phosphor bronze; for the crank pin boxes E and F, with the lighter pressures per square inch of surface and higher surface velocities, we have found nothing so suitable as a lining of genuine babbitt. F has deep flanges extending upward on the sides—broken away in the section—covering the wedge, and acting as a sidewise retainer for B. The upper end of B is also held by shallow flanges on C.

The connecting rod of the Willans' engine, of which there are two on each cross-head, is shown in Fig. 48 for larger powers.

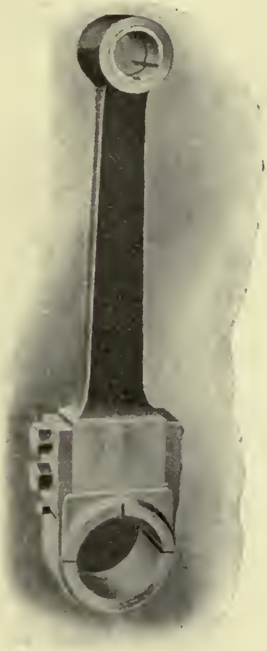


Fig. 46.—Connecting Rod.

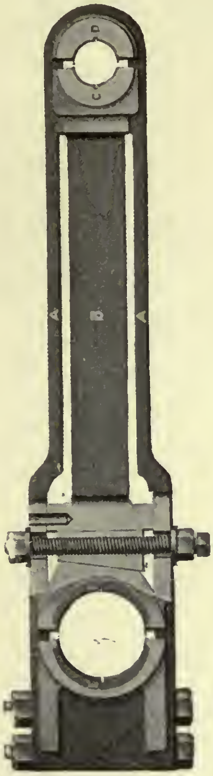


Fig. 47.—Connecting Rod.

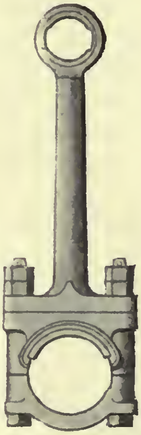


Fig. 48.—Connecting Rod.

The next important elements of the steam engine is the steam inlet and outlet valves. These are of many types, the most common being the slide valve on three ports, as shown in diagram, Fig. 49. It is exceedingly simple and effective, the one valve performing all the required functions; admitting the steam at the beginning of each stroke, cutting it off about half-stroke, exhausting the steam and giving the necessary cushion by compression. The same valve is also employed for reversing the engine, for which purpose certain links are inserted in the gear operating the valve, and connected by a lever by which reversing is accomplished.

Suppose the slide is in the position shown in Fig. 49, and that steam fills the valve chamber, through the steam pipe A. Now, it cannot pass the back of the slide into the upper port D, because the slide is covering it over; neither, for the same reason, can it pass to the exhaust D; but it can pass into the lower port E, in the direction of the arrows, and drive up the piston P, while, as the piston goes up, the steam that drove it down and filled the cylinder on the upper side above the piston is escaping freely through D, in the direction of the arrows, and passing off to the condenser through B, the exhaust port.

When the piston has arrived at the upper end of the cylinder, or at the top of its stroke, the slide V has moved down lower, so that the lower port E is closed against the admission of steam, and the upper one D opened; therefore steam will enter the upper port and escape at the lower, in a contrary direction to the arrows, the piston returning to the bottom of the cylinder.

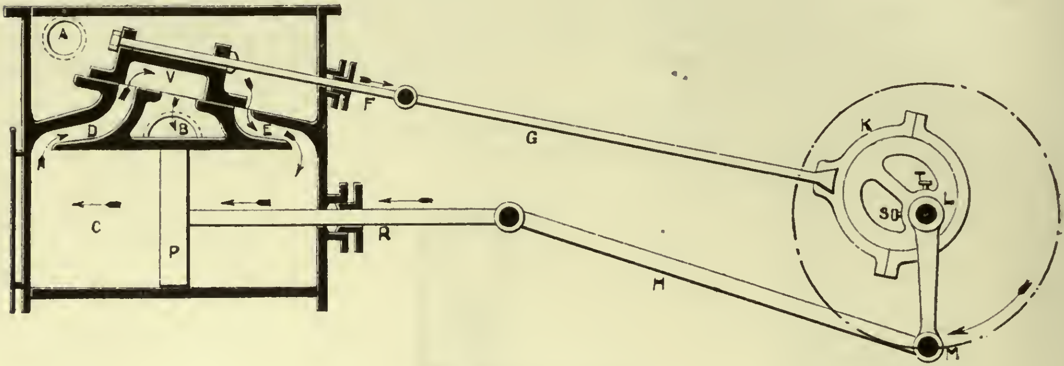


Fig. 49.—Slide Valve.

The slide valve is represented in Fig. 50, in which the shaded part is the slide, and the ports are marked *port*. *c* leads to the condenser. The whole of the drawing is covered over by the slide casing, and steam is brought to the back of the slide at A by the steam pipe (not shown). When the steam is acting it is clearly seen that it presses with great force against the back of the slide at A. The valve rod is shown attached to the back of the slide. When in the position as given in Fig. 50 it is quite evident no steam can pass into the ports and go to the cylinder, as they are both covered over; but when the slide rod moves the valve up the steam can pass into the lower port and drive the piston up, while the steam that is in the upper part of the cylinder can come out at the upper port, when the form of the slide compels it to pass into *bb'* and through *c*, which leads to the exhaust; hence *c* is called the exhaust port. When the slide comes down again both ports are first closed, then the upper one is open to steam and the lower one to the exhaust; precisely the reverse of the first case.

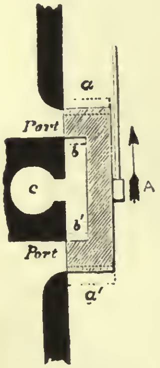


Fig. 50.—Slide Valve.

The "lap" and "lead" of a slide valve are terms applied to the dimensions and relative position of valve and piston.

The width of the opening of the steam ports, for the admission or for the release of the steam at the beginning of the stroke, is known as lead. On the steam side of a locomotive slide it is known as outside lead, or lead for the admission; on the exhaust side it is inside lead, or lead for the exhaust. When the valve is placed at half-stroke over the ports the amount by which it overlaps each steam port, either internally or externally, is known as lap. On the steam side it is named outside lap; on the exhaust side, inside lap. When the terms lap and lead are employed they are understood to refer to outside lap and lead only.

The advance of the eccentric is a term used to denote the angle which it forms with its position at half-stroke, and when the piston is at the commencement of its stroke.

The locomotive slide, as seen in shaded section in Fig. 50, has neither lap nor lead, but if it extended to the faint dotted lines *bb'* it would have lap on the exhaust side to both ports; while, on the contrary, if it reached to the dotted lines *aa'*, it would have lap on the steam side. Lap is chiefly used on the steam side. To see what effect this will have, examine the top port, and suppose the slide going up. It is evident if the slide reaches to the dotted line *a*, as it rises from the bottom of the upper port, it will close it sooner against the admission of steam than it would be otherwise if the slide were constructed simply as drawn in Fig. 50; therefore the steam that has had time to get into the cylinder has to perform the rest of the stroke expansively.

Lap on the exhaust or eduction side *bb'* is always less than that on the steam side, and closes the port to the exhaust sooner than it would otherwise be, and thus prevents all the steam from rushing out to the exhaust: the steam remains behind, and the piston acts against it as against a cushion, and all sudden jar and stoppage is avoided. Sometimes there is no lap, and even less than none, or negative lap; then the valve cannot cover both ports at once. When the slide has neither lap nor lead the breadth of the slide face is equal to that of the steam port, and the travel of the slide twice the breadth of the port; but when the slide has lap the travel of the slide must be double the lap with double the breadth of the steam port.

Let us suppose that, at the instant the piston is at the top of its stroke, the slide is in the position shown in Fig. 50, but that it extends only to the top darkly dotted line, then the port at that instant would be open for the admission of steam; this is what is called the lead of the slide. Remember, the lap is when the slide is at its middle position, but lead when the piston is at the end of its stroke. There is always more lead required in high-speed engines, and it is necessary to open the exhaust early if the clearance spaces are small.

Stephenson's link motion (Fig. 51) is the most common form of reversing gearing. It consists of two eccentrics with rods connecting to link. One eccentric is called "forward," the other "backward," with a slot in it. In the slot moves the block *p*, fastened to the end of the valve rod *a*.

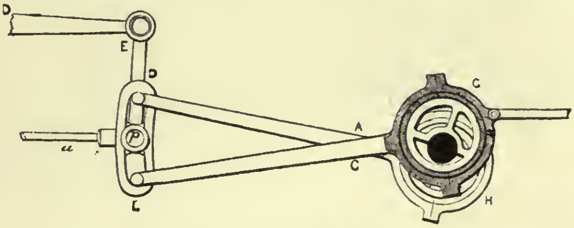


Fig. 51.—Link Motion.

The lever moves the link up or down. When the forward eccentric is moved, so as to work the valve rod, it moves the slide, and the ship or locomotive goes forward; but when the backward eccentric works the slide rod the engine is reversed. The link motion is thus a simple and effective mode of reversing the engine, and without trouble.

If the pin and link be placed in the position shown in Fig. 51 the slide has then but little travel, and we can see that this travel is increased just in the same proportion as the bell crank lever DE*p* moves the link DE up or down from the mid position. As the amount of opening for steam depends upon the motion of the slide, by leaving *p* in different positions in the slot we open and close the port at and during varying times. This is done by not placing the block at the extremity of the link, but at a distance from it, and resting the lever in its proper place. For this purpose an arc or sector with notches in it is attached to the link motion, to fix the handle in and secure the required opening. The slide valve with this reversing gear is a very simple mechanism, and holds its own in all small steam engines. Many valves and gears are more economical, but they introduce complications and refinements altogether out of place in small engines.

The ordinary slide valve is also almost exclusively used on the low-pressure cylinder of large compound and triple-expansion steam engines, as it is less bulky, cheaper, and also more steam-tight than other valves. On the other hand, when used with high pressures and of large size, the friction caused by the immense pressure on the valve faces is very considerable.

The old-fashioned long D valve was an attempt to relieve large valves of this frictional pressure ; in fact, it was the next thing to a piston valve. It was a piston valve with flat faces on one side.

Why it was so made is a mystery, for it would have been cheaper and better as a cylindrical piston valve than as a D section. The piston valve has superseded it entirely. The simplest piston valve is exactly the same in operation as a D valve. This can be readily seen from Fig. 52 of a single-acting engine. It will be quite clear, upon a little consideration, that the valve would operate equally well on one double-acting cylinder as it does upon two single-acting cylinders.

The main valve V is of the piston type, packed with four spring rings, the tightest, most durable, and most perfectly balanced form of valve yet designed. Steam enters through the throttle valve at S, and passes into the annular space around the neck of the valve. In the section through both cylinders the left-hand piston is shown as just starting on its downward stroke, and the right-hand piston beginning its upward stroke. The valve has moved upward by the amount of its lap and lead, and is beginning to open communication between the steam space and the port A, admitting steam to the top of the piston forcing it down. The extreme bottom edge of the valve has uncovered the lower port, establishing communication between the right-hand cylinder and the exhaust chamber E. The crank forces the right-hand piston up, the exhaust passing through the port B under the valve into E and out through the exhaust pipe.

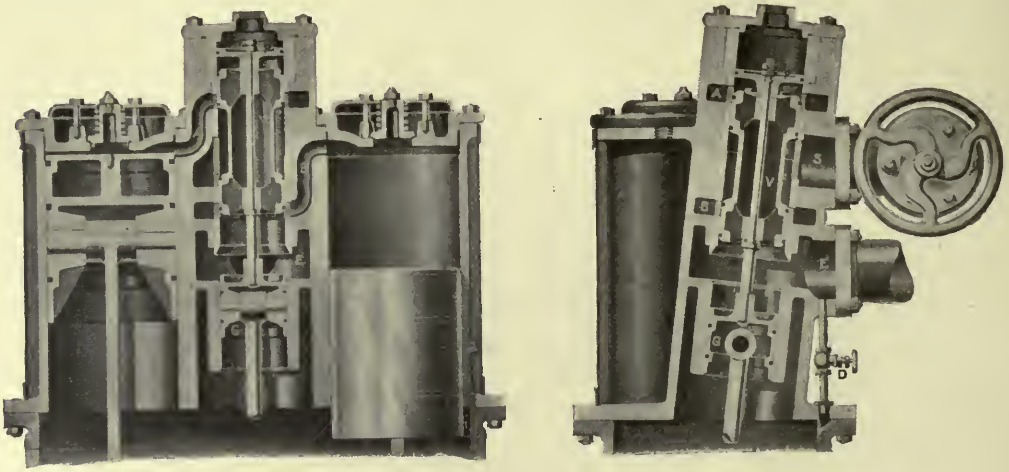


FIG. 52.—Piston Valves.

As the revolution progresses the valve moves upward to the limit of its stroke, and starts downward, covering the port A again at a point determined by the governor, cutting off the steam supply and allowing expansion to take place in the left-hand cylinder. A little later the lower edge of the valve covers the port B, so that the steam remaining in the right-hand cylinder is compressed. When the stroke is nearly completed the upper edge of the valve begins to uncover the port A, when release takes place in the left-hand cylinder. When the revolution is half completed the left-hand piston has reached the limit of its downward stroke, and the right-hand piston is about to begin its downward stroke. The valve is then in the position shown in the right-hand section ; *i.e.* travelling downward, beginning to uncover the port B admitting steam to the right-hand cylinder, and uncovering the port A leading from the left-hand cylinder. During the remainder of the revolution a cycle of operations occurs similar to that just described, except that these operations take place in opposite cylinders. The exhaust steam from the left-hand cylinder passes over the top edge of, and down *through*, the valve, which is hollow, into the exhaust chamber E.

The valve guide G is a small piston of about the same diameter as the valve, which serves as a cross-head for the attachment of the eccentric rod, and at the same time prevents the exhaust steam from passing into the crank case. If the engine were exhausting against any considerable back pressure there would be an unbalanced downward force caused by the pressure of the exhaust over the area of the valve guide G. To provide against this, the steam chest is extended upward, and a back pressure piston P attached to the valve stem, above the main valve, high enough so that it never comes down to the port A. The upper surface of P is open to the atmosphere through a hole in the steam chest cover, while the under side is subjected to the pressure of the exhaust, which exerts a force equal in intensity and opposite in direction to that due to the pressure on G, and consequently makes the balancing of the valve practically perfect for all conditions of both steam and exhaust pressure.

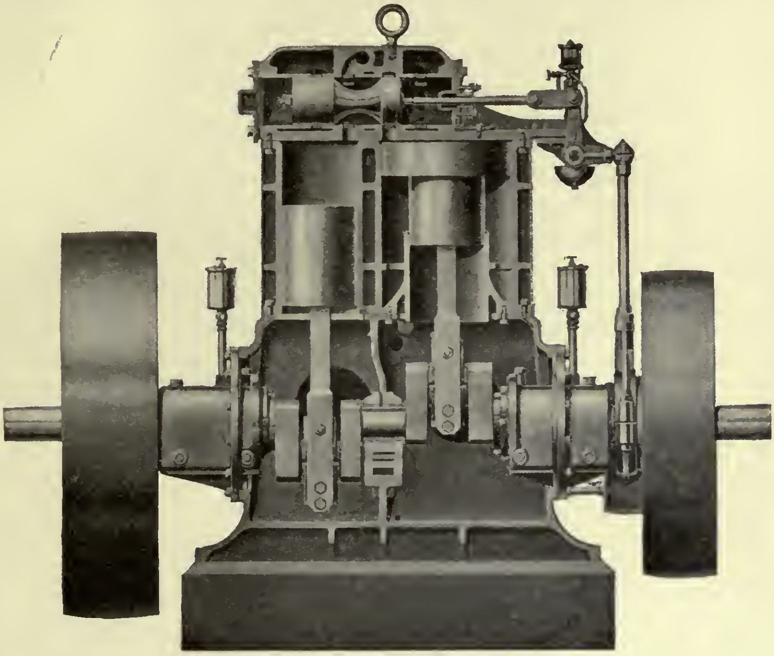


FIG. 53.—Showing Piston Valve.

The adaptation of a simple piston valve to a single-cylinder compound engine is shown in Fig. 53, a section of Westinghouse compound engine. The high-pressure cylinder is at the end of the working stroke, the low-pressure is about to begin the working stroke, and the two cylinders are being put into communication by the narrow waist of the piston valve. This one valve performs all the functions of a compound engine steam distribution system.

The piston valve in the Peache engine is also a good example of a single valve performing many operations. Also the Belliss & Morcom engine valve may be studied with advantage. Perhaps the most remarkable piston valve system is that known as Willans. The valves already referred to all work above, below, or alongside of the cylinders. The Willans valves work inside of the cylinders—in fact, slide inside of the piston rods.

The valves and eccentric are shown in Fig. 54, and the simple 2-cylinder engine in Fig. 55, showing clearly the hollow piston rods and the ports therein.

Piston valves are used, moving *inside a hollow piston rod* 21, which passes completely through the line of pistons, and through the ends of the cylinders. The eccentric is on the crank pin, and not on the shaft, the reason being that the valve face (*i.e.* the inside surface of the hollow piston rod) *moves with the pistons.* Consequently the valve motion required is a motion *relative to the pistons*, and this is obtained by mounting the eccentric on the crank pin, which, like the piston rod, moves up and down with the pistons. Though its lead is set out differently from that of an ordinary eccentric, its effect upon the movement of the valves is exactly the same.

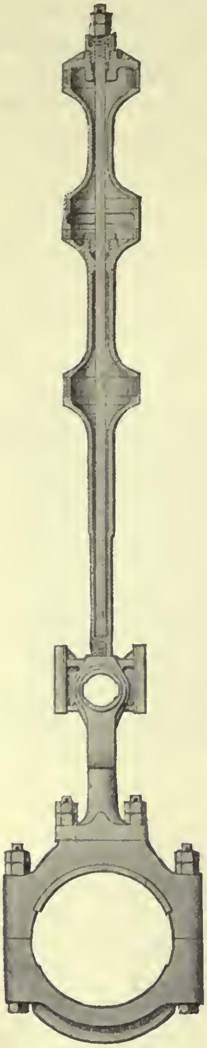


FIG. 54.—Piston Valves.

Starting with a simple engine, the steam passes from the steam chest into the cylinder through the hollow piston rod 21, which moves steam-tight through the gland 11 low pressure in the cylinder top, which forms also the bottom of the steam chest (or receiver, if the engine be afterwards compounded).

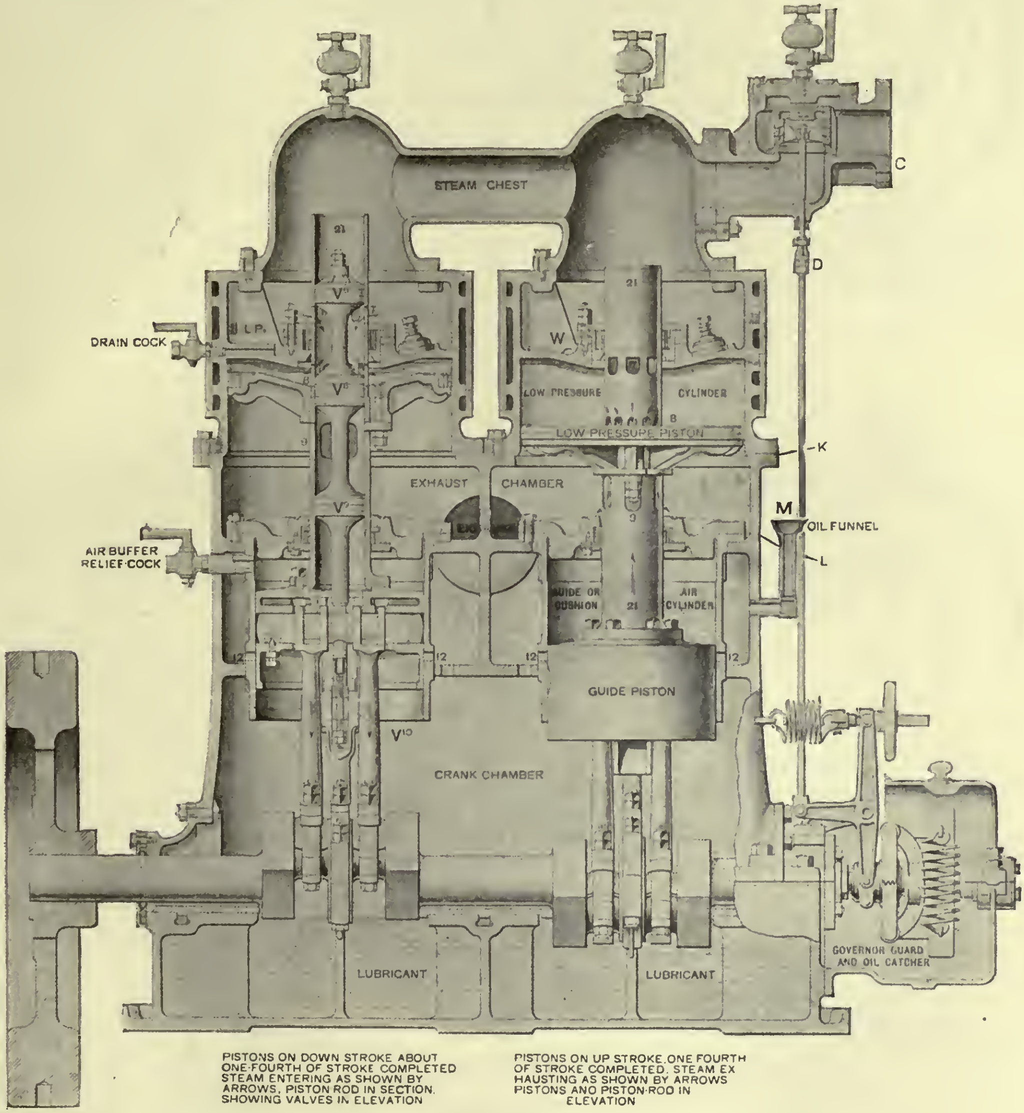
Referring to the left-hand line of pistons (Fig. 55, which has completed one-fourth of the down-stroke), it is seen that steam is entering the piston rod or "trunk" by the ports 7, and passing out again into the cylinder by the ports 8. At about three-quarter stroke the piston valve V^8 will close the last-named ports, but cut-off will take place earlier than this by the ports 7 entering the gland 11 low pressure, and so leaving the steam chest. The point in the stroke at which cut-off takes place is governed by the height above the cylinder top to which the gland rings are raised, by a distance piece placed below them (see W in the sectional view). Normally the ports are cut so as to give cut-off at .6 of the stroke, when the gland rings are as low as they can be fitted; the height to which the rings are raised determines the actual cut-off. The cut-off is fixed with reference to the steam pressure and other circumstances; it may be made variable, when necessary, either by hand or by the governor, but in this case the mechanism employed is different from that shown in the section. When a very late cut-off is required it is effected by the ordinary valve motion; in this case there is no valve piston above that which actually closes and opens the ports, *i.e.* above V^8, V^5, or V^2, as the case may be, the latter being constantly exposed to the pressure in the steam chest.

In the section the piston valve V^8 (which commences to uncover the ports 8 at top stroke) is now at nearly its lowest position relatively to the ports.

Shortly before the piston reaches the bottom of the stroke the valve V^8, then moving upwards, again uncovers the ports 8, and places them in communication with the ports 9, through which the steam commences to pass from the upper to the under side of the piston.

The space below the piston communicates with the exhaust chamber and the exhaust pipe, the steam passing away (through 8, 9, and the exhaust) during the whole of the up-stroke. The exhaust pipe may lead either to the atmosphere or to a condenser.

It will be noticed that the upper crank pin brasses of the connecting rods are wider than the lower ones. This is because the upper brasses alone are intended to be in actual contact with the crank pins; the lower ones are only required in stopping and starting. All the moving parts of the engine are designed to be strictly in "constant thrust"; the connecting rods are always in compression. From the fact that the upper or working brasses never leave the crank pins, and so are never exposed to hammering action, however slight, they exhibit extraordinary durability when properly lubricated;


STEAM CHEST
C
D
21
L.P.
W
DRAIN COCK
LOW PRESSURE CYLINDER
B
LOW PRESSURE PISTON
9
K
EXHAUST CHAMBER
M
OIL FUNNEL
L
AIR BUFFER RELIEF COCK
GUIDE OR CUSHION
21
AIR CYLINDER
12
12
12
12
GUIDE PISTON
V^{10}
CRANK CHAMBER
GOVERNOR GUARD AND OIL CATCHER
LUBRICANT
LUBRICANT

PISTONS ON DOWN STROKE ABOUT ONE-FOURTH OF STROKE COMPLETED STEAM ENTERING AS SHOWN BY ARROWS, PISTON ROD IN SECTION. SHOWING VALVES IN ELEVATION

PISTONS ON UP STROKE, ONE FOURTH OF STROKE COMPLETED. STEAM EXHAUSTING AS SHOWN BY ARROWS PISTONS AND PISTON-ROD IN ELEVATION

FIG. 55.—Willans' Engine. Simple.

at the same time, it is evident that no wear which can take place in them, however great, can lead to knocking, as the connecting rods will follow up the wear automatically. But as the lower brasses should not be too far from the crank pin, the wear should be taken up when it becomes excessive,—say, as soon as it reaches $\frac{1}{32}$ inch, care being used that the lower brasses are not brought actually into contact with the crank pin.

The piston valve of Thom's invention, shown in section in Fig. 56, is one much in use for large powers. Its special feature is the arrangement for producing sufficient compression on the exhausting side of the piston to turn the dead points smoothly, and to economise steam as explained. It consists of two pistons, P_1, P_2, of cast iron, connected together by a cast-iron pipe and fixed to the valve spindle VS as shown. These pistons fit and work in liners L_1, L_2 inserted into the ends of a cylindrical chamber CC, bolted to the side, or to the end of the high-pressure steam cylinder. They are suitably packed with expanding rings R_1, R_2, R_3, R_4, like locomotive piston rings, which are frequently made of bronze. The pistons P_1 P_2 open and shut the two steam ports SP_1, SP_2 from the steam chest, and the exhaust port EP, in the same way as an ordinary slide valve does; but the area of steam port is much larger than that of an ordinary slide valve of the same size across, and there is no unbalanced pressure owing to its cylindrical form, hence no frictional wear and tear except that due to the pressure of the expansion rings R_1, R_2, R_3, R_4. To prevent these expansion rings from springing outwards beyond the bore of the liners L_1, L_2 when they come opposite the steam ports SP_1, SP_2, the faces of the steam ports in the liners are cast diagonally, and so prevents the spring rings from seizing on the edges.

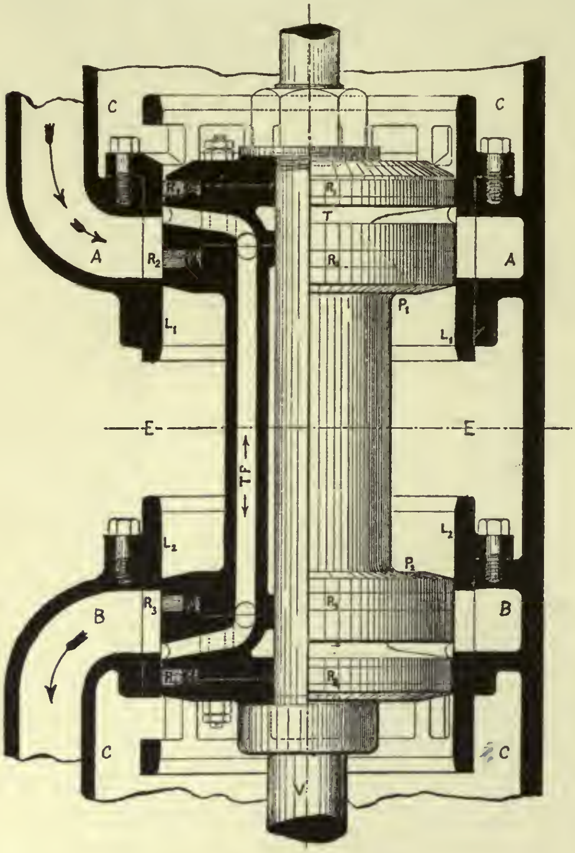


Fig. 56.—Piston Valve.

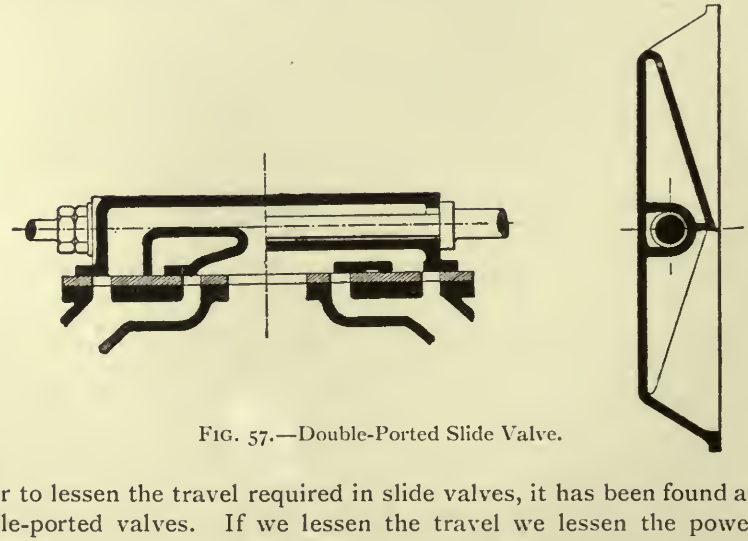


Fig. 57.—Double-Ported Slide Valve.

In order to lessen the travel required in slide valves, it has been found advantageous to use double-ported valves. If we lessen the travel we lessen the power required to move the valve, and also increase the quickness of opening and shutting off steam. In Fig. 57, of a common double-ported valve, it will be seen that there are two steam openings leading to the one steam port at each end of the cylinder, which are opened and closed by the slide valve. The upper opening gets steam from the valve casing direct by its upper edge, and exhausts round a passage at the back of the valve into the exhaust port; the inner opening gets steam from the valve casing through a transverse opening in the slide valve, and it exhausts directly into the exhaust port.

An examination of the sectional view of large engine cylinder (page 23, Fig. 27) will show the application of piston valves.

All slide valves have a gradual admission of steam and a gradual cut-off. It is important that the cut-off should be sudden and sharp in engines with variable loads, and governed by varying the steam inlet, as we have already pointed out in the preliminary survey of the subject; and we have seen the style of valve, called a double-beat cornish equilibrium valve, used for the purpose in the Robey engine.

In this class of valve for steam distribution the valve is lifted from its seat and dropped back again, being held open for a time. The Corliss valve is another form for the same purpose; only, it makes a semi-revolution on a longitudinal axis.

Engines in which Corliss type of valves are used, and also those with drop valves, are usually governed by varying the cut-off, so that their valve gearing is controlled by the engine governor. As it will be better to consider governors and their gearing separately, we will now consider the valves only. Automatic variable cut-off governing can also be applied by combining a drop valve cut-off with an ordinary slide distribution valve, and that often with very considerable economy.

The Corliss gear is very old, but is still able to hold its own in face of all modern improvements.

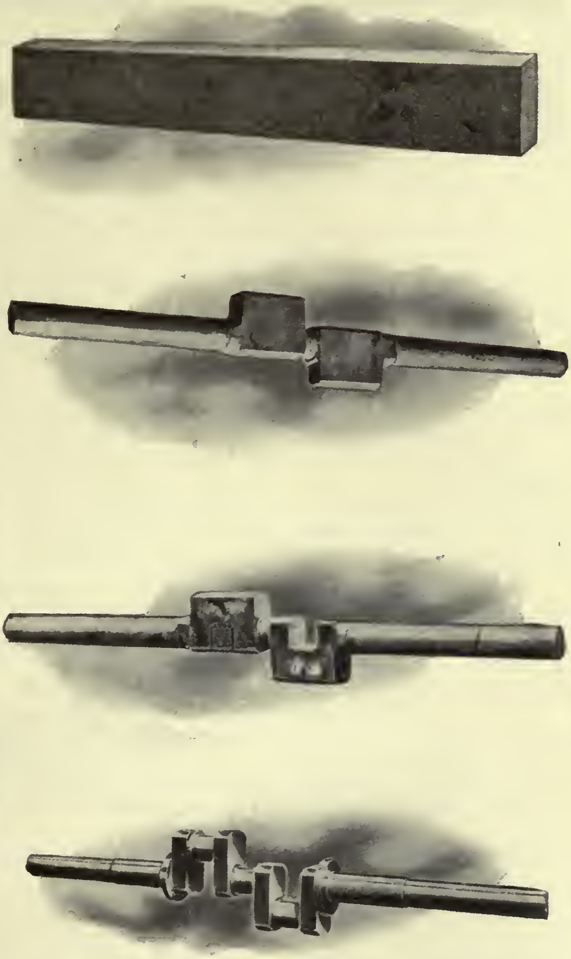


Fig. 58.—Evolution of Crank Shaft from Steel Billet.

The cranks of smaller engines are generally cut out of solid steel. The process is well shown in the four Figures (Fig. 58). First, we have the billet of steel out of which two shafts are forged, so that we get two rough forgings, as in the second. Thirdly, the cranks are rough hewn out by drilling and cutting. The turner and shaper then get to work upon it, and finish it off as in the fourth Figure. This process produces a high class crank shaft without joints, and of homogeneous materials. Large cranks are built up in parts, and these are hollowed out to lessen the weight. The central metal adds nothing much to the strength of a shaft or crank pin, so that it may with advantage be removed. Such a built-up crank is shown in Fig. 59. The crank webbs are shrunk on, and keyed by two keys as well.

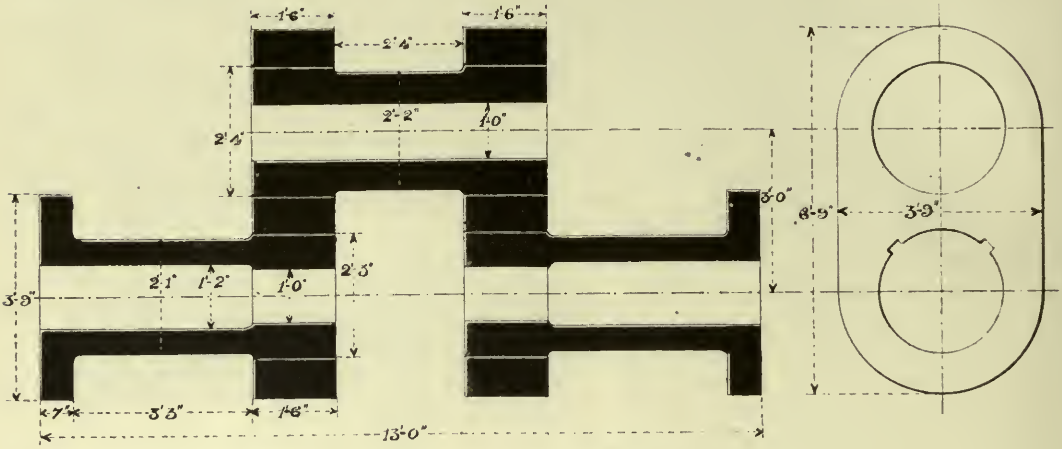


FIG. 59.—Built up Crank for Marine Engine.

The importance of sound and good cranks and shafts cannot be overestimated, especially in marine engines. The stress and strain on huge tunnel shafts is great, and may vary owing to want of alignment, and serious accidents happen through breakage of a shaft.

Lloyds' rules for shafting in marine engineering gives a factor of safety of ten times the torque of the engine, so that it may be ample for any unusual strain.

The failures of tunnel shafting are no doubt due to faulty material. Steel is the best material, but in large ingots it is hardly possible to avoid blow holes. These become long cracks when drawn out. The strains and torque of the driving always tend to develop these faults, and the metal crystallises. If one of these cracks takes a transverse direction the shaft, when subjected to some unusual stress, snaps off. Fig. 60 is from a drawing to scale of the broken end of a shaft.

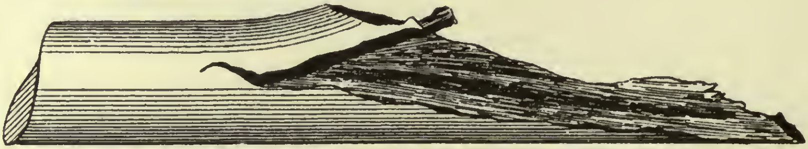


FIG. 60.—Broken Shaft. Scale 1 inch = 1 foot.

The steam turbine which runs at fast speed, and in which the torque is constant, does not strain the shaft so much as reciprocating engines. Smaller shafts are also more likely to be sound in materials.

The adoption of efficient hydraulic jet propellers would, of course, entirely abolish the shaft and external propeller, and this is one of the great advantages of the jet propeller.

REVERSING GEAR.

Single loose eccentric reversing gearing has been used in slow-going engines. It is shown in Fig. 61. The eccentric is an easy fit on the crank shaft, and has a counter-weight L bolted to it. This counter-weight has two stops, against which a feather or key on the shaft comes in order to drive backwards or forward. The operation of reversing is as follows:—First, lift the gab H from the valve rod pin K; second, move the valve V by lever B to admit steam to the end of the cylinder as required for ahead or astern. This causes the key *b* fixed to the shaft to come into contact with the weight stop. The eccentric is then carried round with the weight according to the direction of the crank shaft motion. Immediately the stop comes against the feather or key the gab handle H is dropped and the gab catches into the valve spindle pin K, and the engine will go on. By lifting the gab the engine can be instantly stopped.

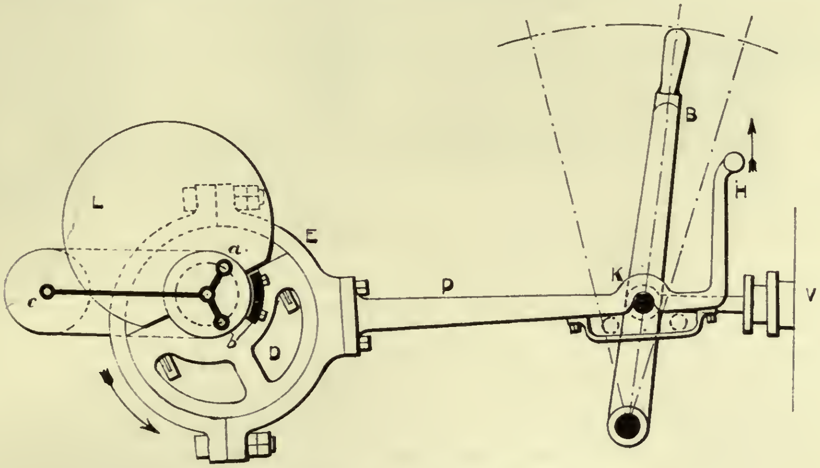


FIG. 61.—Loose Eccentric Reversing Gear.

This gear requires some skill to handle it, even with slow-speed engines. In most cases it is advisable to slow down the engine by throttling the stop valve before commencing to reverse. This gear, with a butterfly valve for throttling, has been used in paddle wheel steamers, and also for colliery winding engines.

JOY'S REVERSING VALVE GEARING

There are some few reversing valve gears differing in design from Stephenson's, inasmuch as they take their motion from the connecting rods instead of from the crank shaft. Joy's valve gear is a good example. It is shown in Fig. 62. Any point on the connecting rod has a motion at right angles to the motion of the piston, and reverses at the moment the piston is at mid-stroke, so that a point K on the connecting rod describes a horizontal ellipse, and a link KP, being attached to the vibrating link PC, causes point L within KP to describe an oval. A third link LF, connected to KP at L, and to the die E, being compelled to travel at its lower end in an oval, moves the die E up and down the curved slot H, point F tracing a true vertical ellipse. The proportions of the links are such that the width of the ellipse exactly equals twice (lap + lead). Now, the curved slot H, whose centre of curvature is D, is carried on a weigh-bar shaft having its axis coincident with the point E. So long, then, as H is vertical, a vertical ellipse is formed by point F, and the valve cannot open more than to lead at each end. This is the "mid-gear" position. Should the weigh shaft be slightly turned, and the slot therefore inclined, an inclined ellipse is formed when B is moved to right or left respectively. The total valve travel would now be represented by the horizontal projection of the ellipses; thus the opening of the steam port would be (the horizontal projection of the ellipse − the width of the ellipse) ÷ 2, measured horizontally, in addition to the lead. When point E crosses the weigh-shaft centre the link takes always the same angular position; the ellipses must therefore intersect at the same points whatever the angle of H. A might have been coupled directly to K, but that an unsymmetrical motion would then have been given to point E. To avoid this error the compensating links CP and PK are introduced.

The shifting of the slotted guide H, so as to incline it from right to left or *vice versâ*, will reverse the engine. Let the crank be moved to the left; the guide being fixed by a left-handed turn, an ellipse will be described, opening the valve to steam at the side of piston required. If the contrary ellipse had been adopted the movement would have closed the valve, or the crank could not have moved in the required direction. It can also be shown that when the guide is constrained to move in the path of the contrary ellipse the crank will turn oppositely to that described.

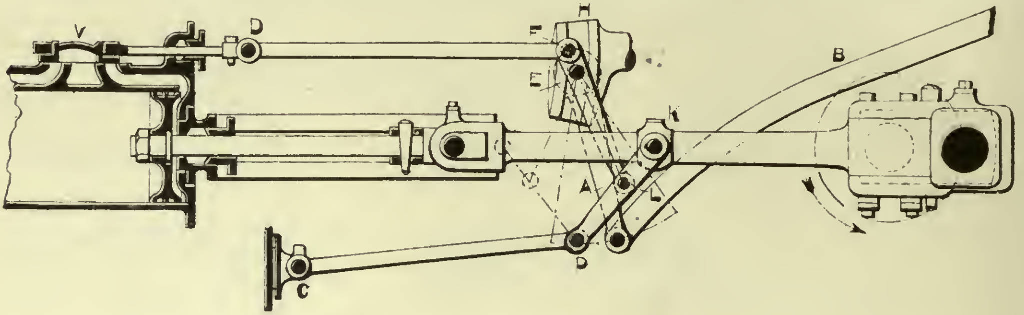


Fig. 62.—Joy's Valve Gear.

THE CORLISS VALVE GEAR

This gear is perhaps the best known. It works by trip gear. The valve is oscillatory, making about a quarter of a revolution between opening and shutting. The shutting is accomplished by a spring, and the gear has a trigger or trip which lets the valve go at the proper moment for shutting.

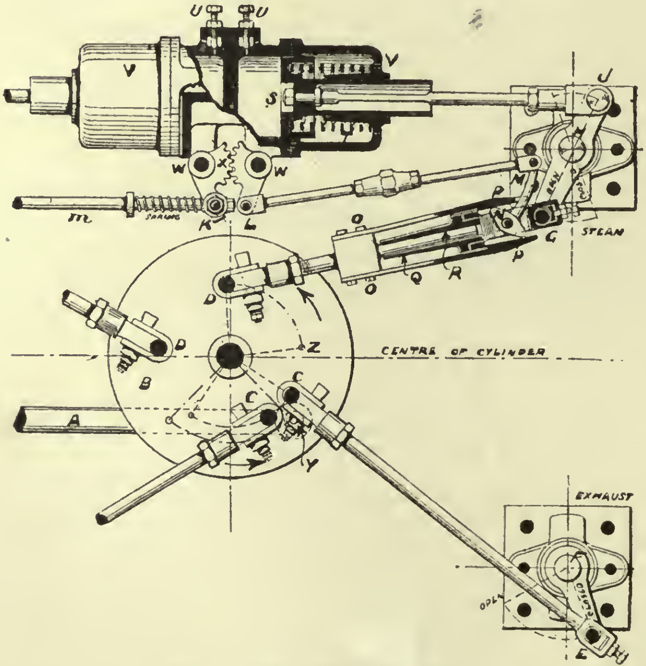


Fig. 63.—Corliss Valve Gear.

The steam valves are close on the cylinder ends, so that clearance is very small. The exhaust valves are distinct from the steam valves, so that there is less loss from condensation. The valves are cylindrical on a shaft, and are cut away to form a passage. Fig. 63 shows the usual valve gear. A is the eccentric rod, which by a to-and-fro motion rotates the disc or wrist plate B; to the latter are connected the four valve rods, two of them at CC actuating the exhaust valves, the other two at DD working the steam valves. The exhaust rod CE is directly connected to the valve lever EF, and moves it through rather less than 90 degrees. The steam valve rod DG is more complicated, consisting of two parts: one, DORP, attached to the wrist plate; the other, QNG, connected to the valve lever GJ. These tend to separate, by reason of the force in the compressed spring T, but are prevented by the spring catches PP. If, however, the latter are prised apart, spring T is released, and, pulling J rapidly to the left, closes the steam valve. The prising action is obtained by the toe lever MN, which, pinned to QNG at N, rocks on fulcrum M. As the pin D moves from Z to D, the rod DG takes a more crosswise position relatively to the toe N, and at some intermediate position the catches PP liberate the parts D and G, permitting the valve to be closed. When D moves back to Z, PP regain their normal condition, and D and G are connected. The position of fulcrum M determines where, between Z and D, the toe shall release part G, and this is decided by the height of governor sleeve, the latter being connected to rod *m*. When the governors rise, *m* is pulled to the left, moving M an equal amount to the right, levers WK and WL being geared together at X. This causes the toe to separate PP at an earlier part of the stroke Z to D, and the converse will happen when the governors fall. Lastly, the dash pot S being full of air only capable of passing out at U, reduces the shock caused by the sudden release of the spring T, the set screws serving to regulate the air passage.

THE PROELL GEAR

This gear is exceedingly simple and effective, and is shown in section in Fig. 64–5. It is a trip gear with drop steam valves and Corliss exhaust valves, not shown, as the under side of cylinder is cut away in the diagram. The lever DE, rocking on fulcrum E, may for the present be looked upon as rigidly connected to the arm FF and toes FG. At point E is attached the eccentric rod, and a movement of D to the right will cause the left-hand toe G, trailing along HJ, to finally slip, when spring L closes the steam valve B. Meantime the right-hand toe which tripped on the last stroke is replaced on the lever JK, by making the lever L free to turn on the pin F until it is high enough to slip in. MM are two dash pots to break the fall of the valves. A rigid bracket SS supports the governor gear; within it the hollow spindle TT revolves, and the balls, flying outward, pivot RT on T, raising the central weight P, while lifting pins VV and the spindle W to a higher position. This rise effects the positions of the toes GG, bringing them nearer together; the reverse happening when the governors fall, and thus is obtained automatically an early or late cut-off respectively. A dash pot within P damps small vibrations on the governor, entrance or exit of air being adjusted by screw Q. The steam valves BB, being double-beat, are balanced. This gear and the Corliss are automatic in gearing, as the governor controls the tripper.

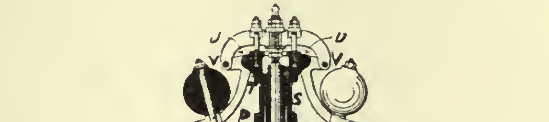


FIG. 64–5.—Proell Valve Gear.

Proell's gear, however, can be applied to ordinary slide valve engines with advantage. The apparatus is shown in Figs. 66, 67, and 68.

A double-beat valve is enclosed in a suitable case, which has two flanges. One of these flanges is secured to the steam chest of the engine, as near as possible to the slide valve, the latter continuing to work in the usual manner, leading the steam to either side of the piston and affecting the exhaust, while on the Proell valve devolves the duty of governing the lead and regulating the cut-off in accordance with the load. The throttle valve is entirely discarded, and the stop valve should always be kept wide open.

Upon the cover of the Proell valve case is arranged a stand, which contains the whole expansion gear and carries the governor on its top.

The Proell expansion apparatus is thus wholly self-contained, embodying the governor, automatic cut-off gear, and the expansion valve, and is sent out from the works finished ready to fit the engine which it is intended to control.

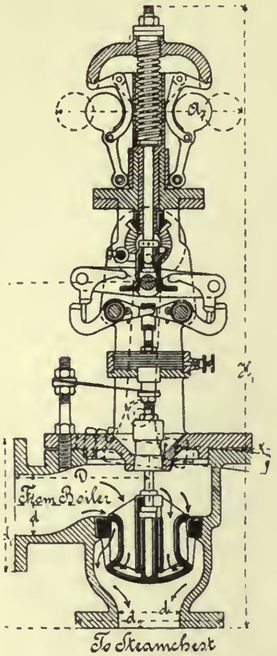


FIG. 66.—Section of Proell's Valve Gear.

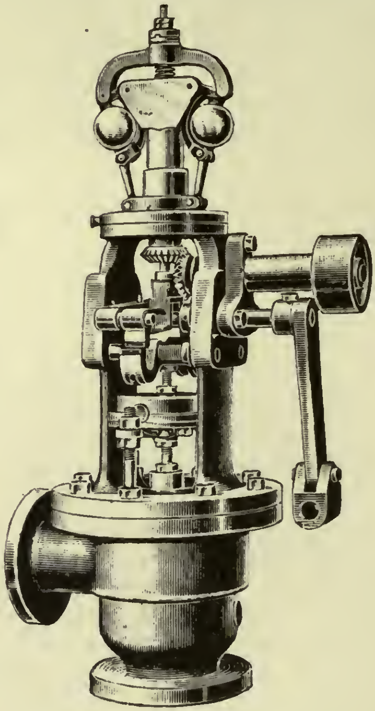


FIG. 67.—Proell's Valve Gear Complete.

Fig. 69 shows the expansion gear proper. It consists of five simple and compact parts—the double armed bell crank lever H, the two bell cranks K_1, K_2, and the two valve lifters L_1, L_2. If we imagine these five parts removed from the apparatus, and connect the governor rod G direct to the valve rod V, then, if the latter controls a suitably shaped throttle valve, the apparatus would be a simple throttling governor. By the mere addition of the above-named five parts we have, however, the most efficient and simple expansion gear.

The governor is a centrifugal spring opposed fly governor to be seen on the top of the apparatus.

To follow the action of the apparatus, Figs. 69, 70, 71, and 72 show the four principal positions and the action of the eccentric rod, hanging arm, gear and valve during one complete stroke of the engine. To follow the motion of the different

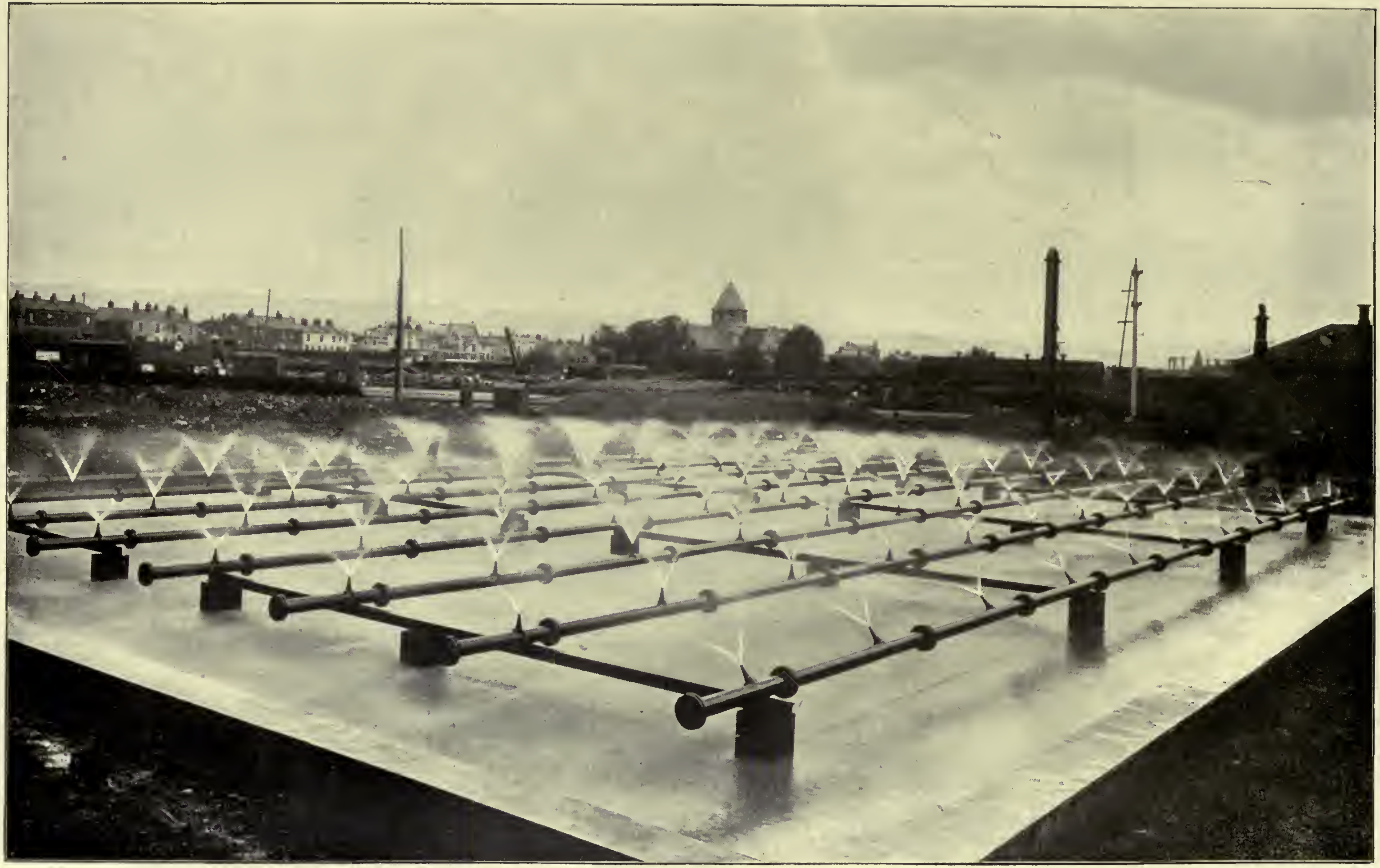


Plate II.—Spray Water Cooling Plant for Condensing Engines.

parts, special attention should be given to the arrows at the bell crank lever, eccentric rod, and valve.

In Fig. 69 the gear is shown in the position which occurs when the bell crank K_2 has just tripped off the lifter L_2 and the valve is closed. The eccentric rod begins to travel from left to right, and the bell crank K_1 descends to meet the lifter L_1. The tail-end of K_1 is raised into the air. As the engine crank approaches the dead centre the bell crank begins to engage the lifter, and the gear comes into the position shown in Fig. 70. The crank is now very near to the dead centre.

In Fig. 70 the bell crank has slightly depressed the lifter L_1, the valve is raised and steam admitted to the steam chest. The tail end of K_1 has already somewhat approached the tripping fork, but is still quite free of the same. This is a very important point, as at this moment the heaviest work of the gear—the opening of the valve—has to be overcome. None of this work can react upon the governor, as the latter is entirely out of contact.

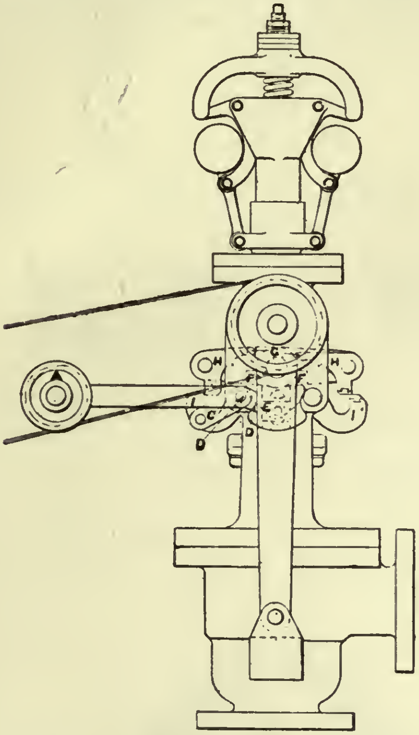


FIG. 68.—Proell's Valve Gear, fitted with Safety Pulley.

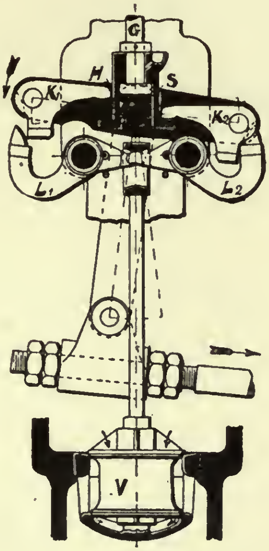


FIG. 69.—Action of Proell's Gear.

In Fig. 71 the bell crank has come down still lower; the valve is entirely open, and consequently perfectly balanced, and it is only now that the governor comes into play: the tail end of the bell crank comes into contact with the tripping fork, the steel faces of bell crank and lifter leave one another, the valve drops and cuts off the steam as shown in Fig. 72.

The gear is now in the same position for the return stroke as it was in Fig. 69 for the stroke just described, and the same action is repeated at every stroke of the engine.

The two valve lifters are independent in their movements, as is apparent. While the lifter L_1 is at work, L_2 is kept at rest by a light spiral spring round its shaft, and lies with its inner end on a rod inserted in the stand.

In Figs. 69 to 72 only the acting parts of the gear are shown. From Fig. 66 it will be seen that a flat spring is provided to assist a sharp cut-off; a dash pot and air buffer serve to cushion the fall of the valve.

Action of the Governor.—From the foregoing description it will be seen how simply, yet directly and reliably, the governor controls the duration of the opening of the valve, in other words, regulates the expansion automatically. The higher or lower the governor holds the tripping fork, the sooner or later will the tail ends of the bell cranks come into contact with it and release the valve.

The governor is generally belt driven, and the belt may break, in which case the engine would run away and might cause a fracture of the fly-wheel if not heavily loaded at the time. A safety arrangement (Fig. 68) is used to meet this emergency, a jockey pulley carried by a lever rides on the belt, and so long as the belt is whole the lever is held up, but should the belt break the lever falls with the pulley. The lever has a short toe under the governor rod, which lifts the rod and the tails of valve trippers so that they miss the valve levers and the engine then stops.

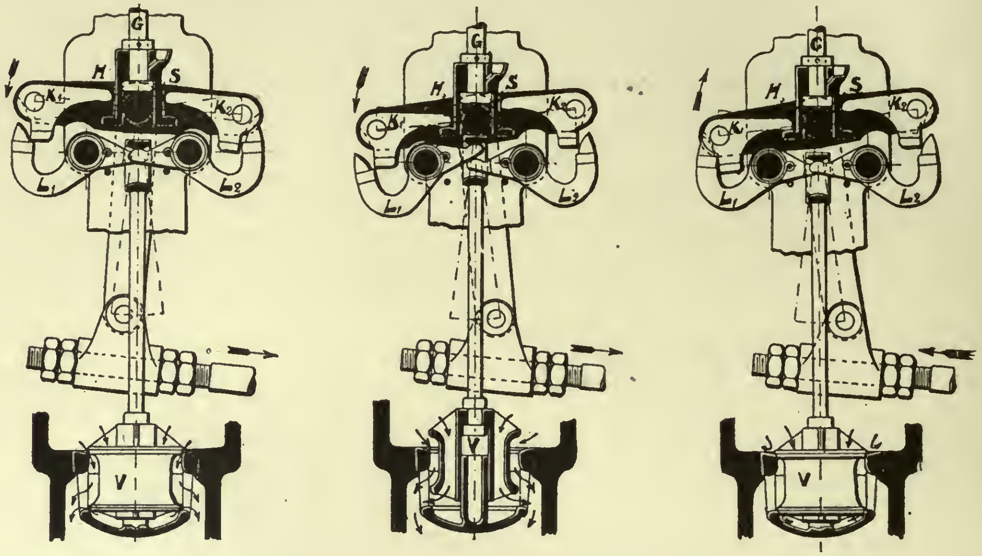


FIG. 70. FIG. 71. FIG. 72.

Action of Proell's Gear.

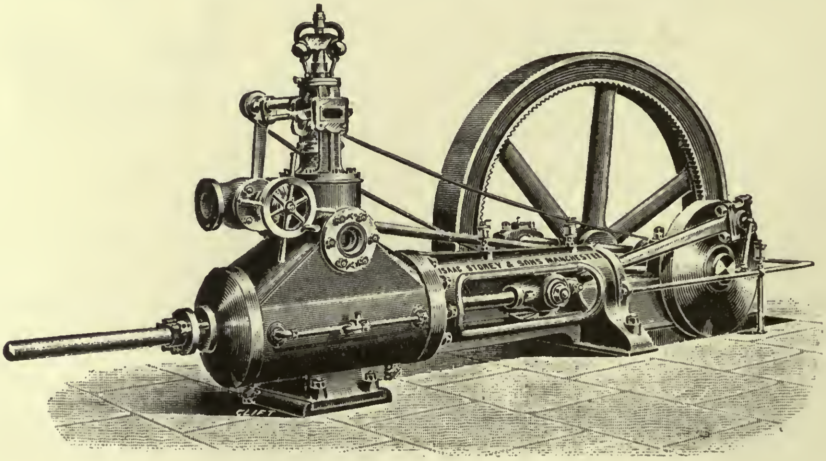


FIG. 73.—Engine with Proell Valve Gear.

Fig. 73 shows the apparatus fitted to a horizontal slide valve engine, as made by Messrs. Isaac Storey & Sons, Manchester.

THROTTLING GOVERNORS

Watt first designed a steam engine governor on the principal of a centrifugal pendulum. The governors of this class have two principal parts, the centrifugal and the valve. Watt's centrifugal is shown in Fig. 74. It consisted of two heavy balls W hung on arms which were hinged at A on a pin, two links EE attached at mid-position to pins BC, and to a sliding collar E; when the balls revolve they fly out, and obviously the sliding collar will be lifted upwards. The higher the speed the farther the balls fly out, and the greater is the lift of the collar. If by rods or otherwise we connect a valve in the steam pipe to the collar E, it is clear that by this centrifugal we could arrange matters so that as the speed increased the valve would shut down, and that at normal speed the valve would be about half open. This governor would then open wider the valve when a load came on to slow the engine down, and close the valve a little more as the load went off. Such a governor could not possibly keep up a constant speed, it depends for its action on the fact that the speed will be inconstant. The two balls, it will be observed, were acted upon by two forces, centrifugal force and the down pull of gravity. The centrifugal force sends the sliding collar downwards and gravity sends it upwards. Now, in order to get sufficient power to work the valve the balls must be heavy; and further, in order to make the lifting and depressing power on the collar equal, the speed of revolution had to be slow. The speed of the engine had to vary considerably before the governor had any effect in opening or shutting the steam valve. This primitive arrangement is still to be found in many steam engines.

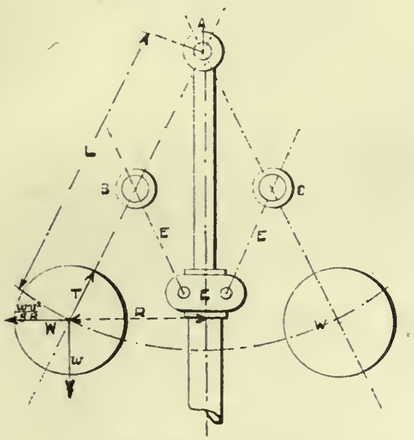


FIG. 74.—Watt's Governor.

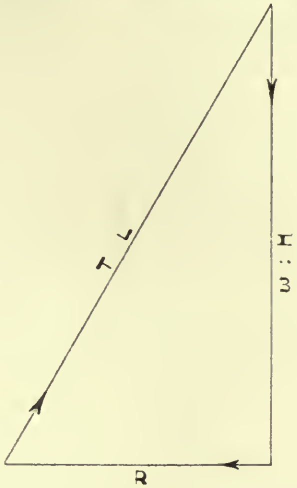


FIG. 75.—Governor Forces.

A force diagram as shown in Fig. 75, in which the three forces acting in a pendulum governor are represented by L, R, H (which are respectively proportional to T, $\frac{wv^2}{gR}$ and W), in which L is the length of the arm carrying the ball measured from centre to centre, $\frac{wv^2}{gR}$ gives the centrifugal force. R is the radius from centre of spindle to centre of ball, and H the height of the pin above centre of ball vertically.

Now, as $H : R :: w : \frac{wv^2}{gR}$, and $H = \frac{gR^2}{v^2}$. But $v = 2\pi Rn = \frac{2\pi RN}{60}$, $\therefore h = \frac{gR^2}{v^2} \times 12 = \frac{gR^2 \times 60 \times 60 \times 12}{4\pi^2 R^2 N^2} = \frac{35200}{N^2}$ inches. And $N = \sqrt{\frac{35200}{h}} = \frac{18766}{\sqrt{h}}$ revolutions per minute, or $h = \propto \frac{1}{N^2}$, and $N = \propto \frac{1}{\sqrt{h}}$, in which h is the lift or motion of the collar due to n the variation of speed, and N is the speed in revolutions per minute.

In order to obtain a more sensitive governing centrifugal it is necessary to introduce another weight or force so as to separate the two functions, the lifting by centrifugal force and the opposing force to open the valve. This is done in the Porter governor by a heavy weight added to the sliding collar as shown in Fig. 76. The balls may be made small and revolved rapidly, so as to obtain working power. The first thing to notice in this design is this, the balls may rotate up to a certain speed without lifting the weight at all, hence the valve is not moved until the speed rises sufficiently to give the small balls enough force to move the heavy weight. And again, it will be seen that any slight increase of speed beyond that required to move the heavy weight upwards will lift it up a great distance.

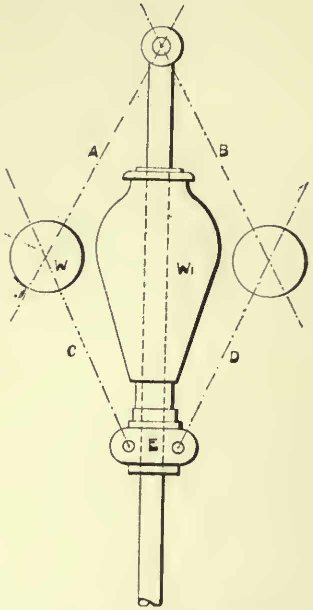


FIG. 76.—Porter Governor.

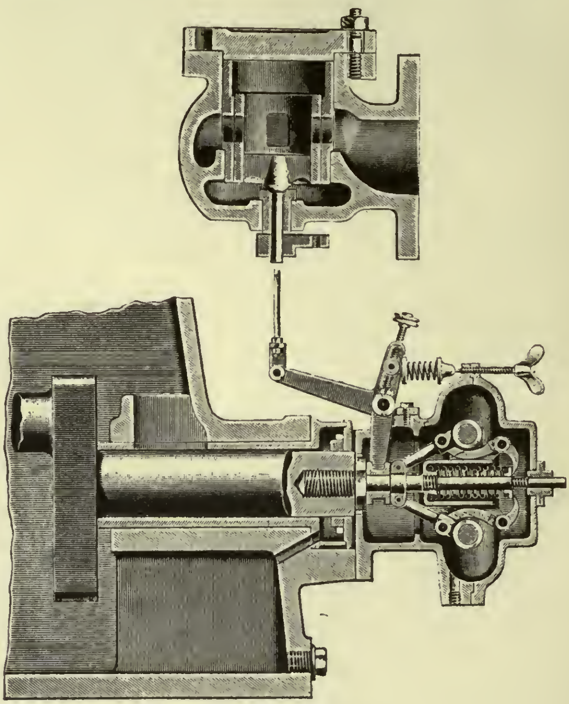


FIG. 77.—Spring Governor.

The four arms are equal in length. The rise of weight on the collar is twice that of the small balls.

With a variation in speed from N to N^1, the height varying from h to h^1, the difference in revolutions would be equal to

$$\sqrt{\frac{w}{w + w_1}}\, 187{\cdot}6\left(\frac{1}{\sqrt{h}} - \frac{1}{\sqrt{h_1}}\right)$$

w being the weight of the small balls, and w_1 that of the weight on the collar.

In some of these governors the weight of the small ball is only 3 or 4 lbs., while the large one is 250 lbs.

Instead of weighting the collar a spring may be applied to oppose the centrifugal force, and thus make the governor lighter altogether in weight. A governor of this class is shown in Fig. 77 fixed direct to the engine shaft and enclosed in a box. The bell crank has an extension and a screw for adjusting an external spring by hand. By this means the speed of the engine can be adjusted without stopping the engine. The valve is clearly shown in section in Fig. 77

Perhaps the best governor of this type, and withal the simplest, is the Pickering governor. In this governor, shown in its various forms in Fig. 78, the balls are carried on laminated flat steel springs, thus doing away with pin joints. On all other governors there are from four to eight pin joints, all subject to wear, get slack, and give trouble. In this governor no trouble from joints can occur. It is a governor which really governs if its speed is properly adjusted. It is, moreover, a fine example of a good invention, being neglected for many years, while far more elaborate and expensive governors are being made and sold by hundreds, although in most cases they were not capable of approaching this governor in their governing powers. It was never heard of until electrical engineers found it out. It is an old American invention of great value.

On the whole question of governing engines the engineer should carefully study each particular case. Many engines are fitted with elaborate governors at great expense which do not require governing at all. An engine to run at a constant load requires only a simple governor to provide against an accidental increase of speed. Many mill engines run week in week out all the year round with the same load, and yet nearly every one is fitted with an expensive governor, which may be ornamental but is certainly not useful ; in such cases automatic expansion is useless also.

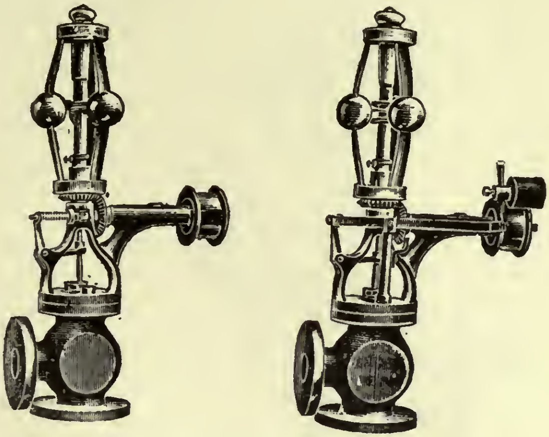


FIG. 78.—Pickering Governors.

Where the load varies not more than 25 per cent., a Pickering or other simple governor is all that is required. Where the load varies more than this, then automatic expansion governors are to be preferred.

PROELL'S SPRING GOVERNOR

This governor (Fig. 79) consists of two inverted conical pendulums, which are suspended by bell cranks from ears cast on the spring box. The points on which the stems of the balls are pivoted are usually chosen outside the centre lines of those stems. The result of this exceedingly simple mechanical combination is that the centres of the balls on opening describe approximately, and as nearly as is requisite, a plane. By means of bell-crank levers at the top of the governor the centrifugal force of the balls is counteracted by a very powerful spring contained in a case, which is securely attached to the central spindle of the governor.

Whenever the governor is rotated at a predetermined speed the centrifugal power of the balls overcomes the resistance of the spring, the balls open, and the sleeve of the governor rises.

An apparently analogous action is claimed for nearly all spring-loaded governors. There is, however, one vital difference between them all and the Proell governor. While the centrifugal weights of all other governors on opening describe an arc about some fixed pivot, the balls of the Proell spring governor open in a straight line at right angles to the central spindle of the governor. This apparently slight difference accounts for the perfection of regulating obtainable with the Proell spring governor, unequalled by the performance of any other.

It is a physical law that the power exerted by a spring increases at an equal ratio to the compression to which it is subjected. To make a spring-loaded governor so regular that, irrespective of its degree of opening, every variation in speed shall produce the same variation of angular opening, it is indispensable that the centrifugal force of the balls shall increase or decrease at the same ratio as the compression of the spring.

This is the case with the balls of the Proell spring governor, where the centrifugal force varies in the proportion of their radial distance from the centre line of the governor, as well as with the angular opening of the hanging straps. It cannot be the case with a governor the balls of which open in an arc.

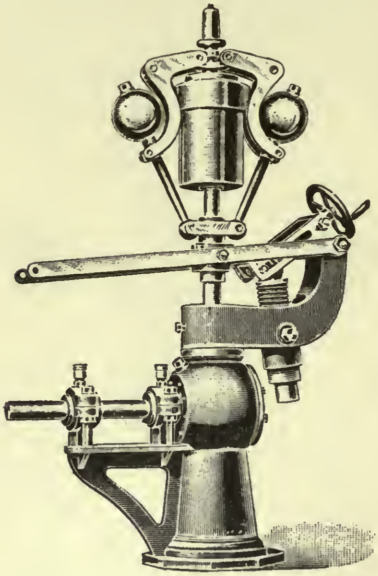


FIG. 79.—Proell Spring Governor.

The following example will show the correctness of this contention. Suppose a spring governor constructed in the following manner. It has two ordinary conical pendulums, 11 inches long, with balls at their ends weighing 8 lbs. each; the upper ends of the pendulums are provided with thumbs pressing upon a spring, which at two inches compression exerts a power of 120 lbs. If such a governor were rotated at a speed of 178 revolutions per minute the centrifugal force of the balls would just balance the spring power; if the speed were then increased to 180 revolutions, the increase of centrifugal force would compress the spring a further $\frac{5}{8}$ inch and open the balls 10 degrees, that is, with an acceleration of not quite 1 per cent. In order to open the balls a further 10 degrees it is necessary to increase the speed to 190 revolutions per minute, owing to the balls being guided in an arc, while the power of the spring increases at the same ratio as for the first 10 degrees. Assuming that this opening of 20 degrees corresponds with the friction and full loads of the engine, it follows that with the former the speed must be about 7 per cent. higher than with the latter. The degree of irregularity would be 7 per cent. This is, however, a favourable instance of regulating by a spring governor of the "Arc" type; there are specimens in the market whose degree of irregularity is as great as 17 per cent.

The table below shows the power and regulation of this governor.

	Number of Governor.					
	I.	II.	III.	IV.	V.	VI.
Revolutions per minute	190	180	170	160	140	120
Total lift of the sleeve	$1\frac{13}{16}$ in.	$2\frac{1}{4}$ in.	$2\frac{11}{16}$ in.	$3\frac{1}{8}$ in.	$3\frac{5}{8}$ in.	$4\frac{1}{2}$ in.
Regulating power in the sleeve, at 2 % variation	$3\frac{1}{2}$ lbs.	$6\frac{1}{2}$ lbs.	13 lbs.	22 lbs.	31 lbs.	48 lbs.
Energy of governor in foot-lbs.	$\frac{1}{2}$	$1\frac{1}{4}$	$3\frac{1}{4}$	$5\frac{3}{4}$	$9\frac{1}{4}$	18

The automatic cut-off fly-wheel governor operates by shifting the eccentric so that it has more or less eccentricity, thus altering the amount of steam admitted. As an example of this class of governor we may examine the Sentinel engine governor shown in Fig. 80, while Fig. 81 shows a series of diagrams in small high-speed engines.

The highest economy at varying loads is obtainable by the use of this device (which in its essential features is old and well proven), as by its action the amount of steam supplied to the cylinders is proportioned to the power given out. A further economy is made at light loads, as the valve cutting off the exhaust before the end of the upward stroke causes a certain amount of steam to be compressed at end of upward stroke. This steam, expanding on the downward stroke, gives back the work expended in compressing it, while it has during the interval, by occupying the clearance space at the head of the cylinders, saved the waste of the live steam necessary in an ordinary throttling engine to fill such clearance each time, and by the heat induced by compression saved a considerable amount of the initial condensation, which is a fruitful source of loss. The line drawing shows construction of governor, and the indicator diagram, which is a reproduction of one taken from a stock 8½-inch by 7-inch engine, shows its effect on the distribution of steam. The lines, full and dotted, which are lettered similarly in capitals and small letters, represent the varying pressures of the steam in the cylinder during the downward and upward movement of the piston respectively, the load being increased from nil to 40 electrical horse-power by increments of 10 horse-power.

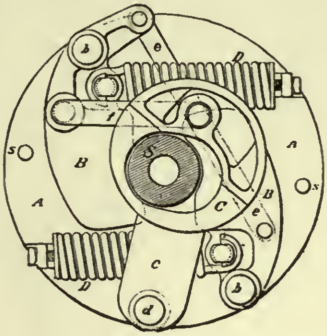


Fig. 80.—Fly-wheel Governor.

The action of this governor is instantaneous, and within 2 per cent. variation from the mean speed when full rated load is suddenly thrown off and on. No engine which does not govern within this is sent out. This is not, of course, extraordinary. The friction is always the same owing to its running all the time in an oil bath. Again, this governor maintains the speed stroke by stroke at the rate to which it is set, and not merely at a uniform average speed counting minute by minute. Nothing but a complete bathing of the governor in oil will accomplish this close governing.

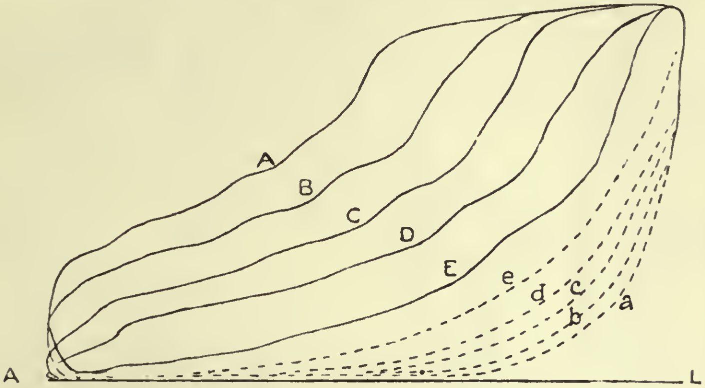


Fig. 81.—Diagram showing Automatic Variable Cut-off Governing.

The value of this feature in dynamo driving, for instance, will be apparent to all electricians who appreciate the great waste of incandescent lamps caused by constant small fluctuations of the electromotive force.

ENGINE VALVES

These are principally the slide or other valves on the cylinder for steam distribution and cut-off, and have been noticed under Governors. There is also the stop valve, which is of some importance with high-pressure and superheated steam.

The governor valve when throttling only is usually a butterfly or double-beat equilibrium valve, so that it can be easily moved.

In diagram, Fig. 82, the common butterfly valve is shown; it was the original valve designed by Watt for his governor. The valve is an elliptical door fixed upon a spindle which passes through a section of the steam pipe. The spindle passes through a packing gland to levers, whereby the valve may be opened or shut or partially shut.

Usually in steamships there is a throttle valve of this kind between the cylinder and the stop valve, as it is a very simple and handy valve for quickly checking the speed of the engine. The stop valves are usually screw-down valves requiring some time to close them, whereas the butterfly valve can be closed at once by a lever. They are not steam-tight, hence cannot be used alone as stop valves.

An equilibrium double-beat valve is shown in Fig. 83, much used for governor valves, for which purpose the two valves do not come down against the seats but slide into them when shut. They do not entirely cut off steam in that case, but reduce it sufficient for governing. When full open the valves are clear above the seats, allowing full bore of steam to pass.

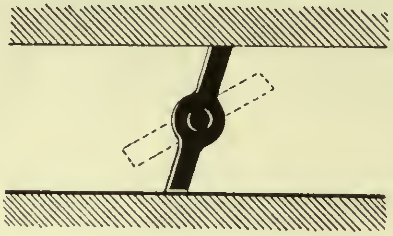


Fig. 82.—Butterfly Valve.

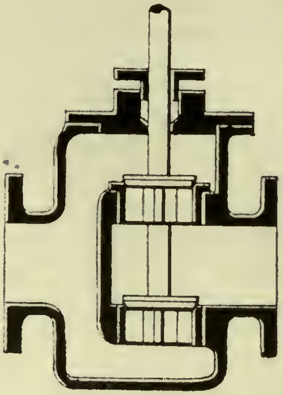


Fig. 83.—Equilibrium Valve.

A somewhat similar valve is shown in Fig. 84, the Cornish valve. It has two seats, so that it is balanced. This valve is used in the Proell engine gear, also in Robey's engines and others with trip gear.

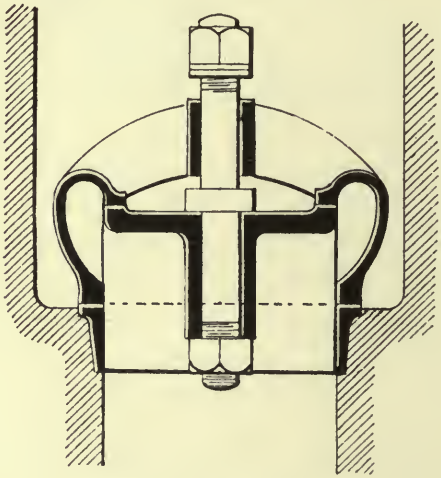


Fig. 84.—Cornish Valve.

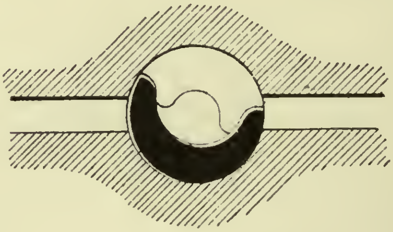


Fig. 85.—Corliss Valve.

The Corliss or oscillating cylindrical valve is shown in Fig. 85. Sometimes these are conical in order to keep them steam-tight.

GARVIE'S VALVE

This valve is made by Messrs. W. H. Bailey & Co., Salford, for steam pressures up to 200 lbs. per square inch, and is shown in Figs. 86 and 87. The discs forming the inner gates are parallel to each other, and slide between parallel valve seats. The discs have between them a circular supporting plate, through which the spindle for opening and closing the valve passes, but they are free to rotate on this supporting plate, so that they may adjust themselves in order to ensure that the junction between the valve and seat may always be tight; and to this end also springs are placed between the valve discs and the supporting plate, as shown on the section in Fig. 86. The discs and the body of the valve are fitted with detachable rings of hard gun-metal, which are secured to their places by screws; but in addition to the screws an annular rib formed on the discs and on the body of the valve engages with an annular groove in the ring, but without fitting tightly therein, so that the differential expansion and contraction of the disc and ring by reason of variation of temperature may proceed without straining the metal and warping them. To enable the rings which are fitted to the body of the valve to be readily removed and replaced, the body is made in two parts, as shown in the cross section. Each valve is tested by hydraulic pressure to

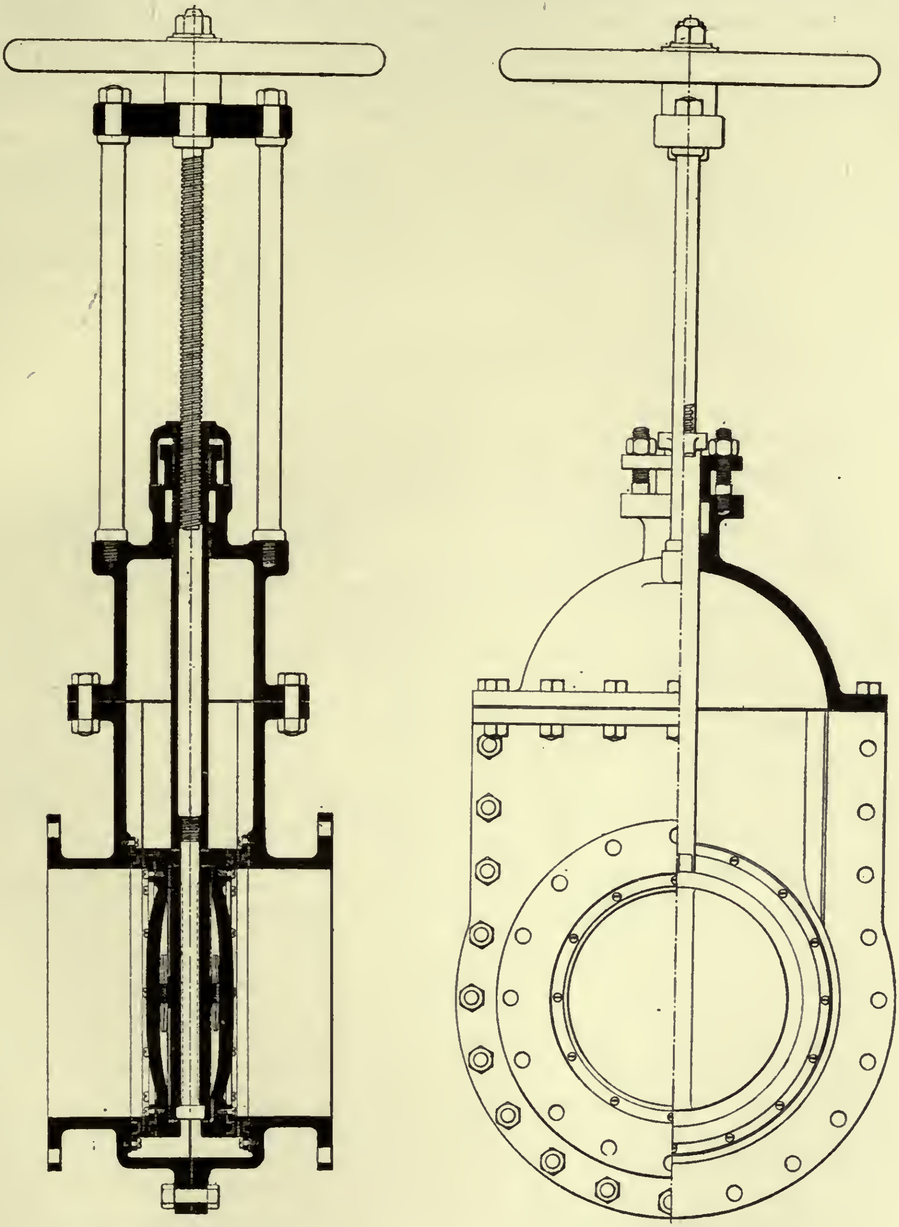


FIG. 86.—Cross Section. FIG. 87.—Half Section.

Garvie's Valve.

450 lbs. and by steam to 250 lbs. per square inch. The 2-inch, 2½-inch, and 3-inch valves are of gun-metal, the 4-inch size of gun-metal or cast iron, with gun-metal fittings, and those of 4-inch and upwards, are of cast iron or steel.

A novel combination of great utility is shown in two sections in Figs. 88 and 89. It is a valve of the Hubner & Meyer's patent type, which is being introduced into this country by T. Sugden Limited, 180 Fleet Street, London, E.C. The chief feature of the valve consists in its closing automatically in both directions in case of accident—that is, either in the event of a fracture or other mishap to the steam pipes or engines on the one hand, or on the other hand, in case of any accident or failure in any part of the boiler. If a tube should burst in a water-tube or multitubular boiler, for instance, the valve automatically closes and prevents any escape of steam from the boilers at work through the defective boiler. The valve also acts as an ordinary stop valve. It is worthy of note that more lives are lost owing to what may be termed minor accidents than result from actual boiler explosions. A larger percentage of such accidents result directly from scalding, owing to the escape of steam and water, which makes it difficult, and in many cases impossible, to approach the stop valve. For this reason it is desirable to take all possible precaution to ensure safety. Another feature of the valve is that it automatically connects the boiler in which steam is being raised with the boilers at work and under full steam. The steam pressure on the outlet or upper side of the valve keeps it closed until the pressure in the boiler being set to work is sufficient to open it. Boiler attendants are at times deceived by inaccurate steam gauges, and are liable to turn on steam and connect the boiler when there is a considerable difference in pressure. Many accidents have resulted from this cause.

The valve is constructed to work at 300 lbs. pressure on the square inch, the bodies being made of cast steel having a tensile strength of about 30 tons on the square inch, with an elongation of 20 to 25 per cent. in pieces 4 inches long. The working parts are made of a special mixture of gun-metal, and the seating is of a special combination of nickel to withstand the action of superheated steam. In order to protect the valve itself from any liability to fracture it is made excessively strong, and for this reason cast iron is entirely avoided. Hand wheels are provided for testing and operating the isolating valve from the outside, and the valve spindle is so arranged as to cut out the stuffing box when the valve is fully opened, and thus protect the packing from the action of the steam. This arrangement also admits of new packing being put in while the valve is under full pressure.

Fig. 88 is a section of the valve shut, and Fig. 89 a sectional view of an angle stop valve open. The horizontal flange A is connected to the boiler, and the vertical flange B to the steam pipes. The arrows indicate the direction of the flow of steam. The combined valve consists of two separate valves—the upper valve C and the lower valve D, both being loose on the spindle. When valve C is screwed down it acts as an ordinary stop valve and shuts off the boiler; and when the spindle is raised the valve is opened by the flow of steam from the boiler, when the pressure on both sides is equal. The valve C, being loose on the spindle, acts as a non-return valve, and immediately closes in case of accident to the boiler, and prevents escape of steam from the other boilers. The lower valve D operates in the opposite direction,—that is, in case of steam-pipe fractures or any other mishap on the outlet side of the valve. Under normal working conditions this valve is kept in position by its own weight and a certain pressure of steam caused by a partial vacuum formed by the flow of steam past the valve, which exerts a suctional action in the space between the valve and spindle, as indicated by arrows at E. The valve can be set to close at any desired difference in pressure by an adjusting screw. When once adjusted a padlock or seal may be applied to prevent anyone tampering with the valve. When an accident has occurred and the automatic valve D is closed against its seat T, it is kept in this position by the steam pressure until the stop valve C is closed. When this is done the steam pressure on both sides of the valve D is equalised by a small quantity of steam passing to the upper part of the valve through the space between the valve and spindle; this causes the valve to return to its normal working position.

Hand wheels for testing are connected by means of spindles to the levers F and G, and in this way the valves C and D can be controlled from the outside and kept in working order.

A valve facing H is formed on the spindle, which, when the steam stop valve C is fully open, comes in contact with the seating J immediately under the stuffing box; this protects the packing, and also admits of new packing being put in while the valve is under full steam. The working parts of the valves C and D are made of a special mixture of gun-metal, and the seating K is made of a special preparation of nickel, for working with superheated steam.

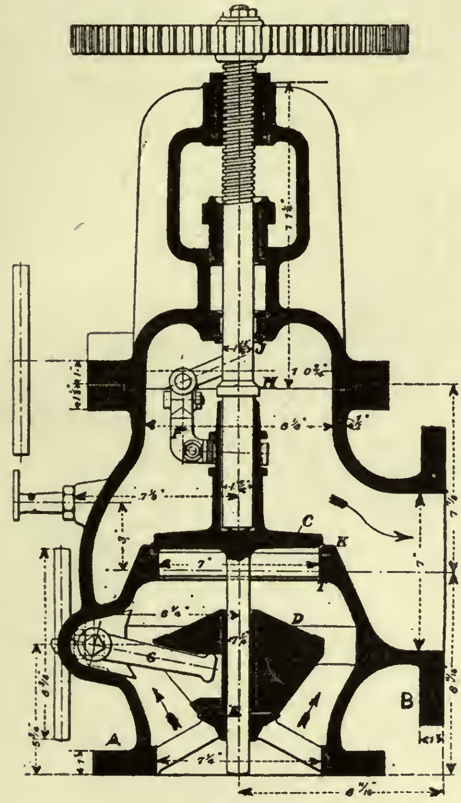


Fig. 88.

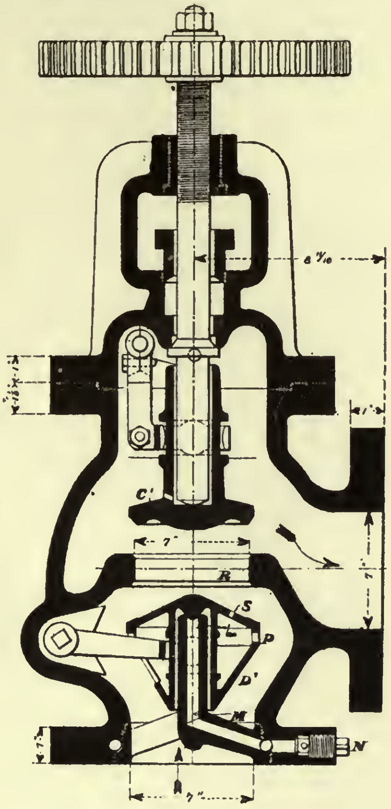


Fig. 89.

Meyer's Stop Valve.

Fig. 89 illustrates the type of valve used for marine purposes, as well as for land purposes, when it is desired to place the valve at an angle or in an inverted position. This valve is in all respects the same as shown in Fig. 88, excepting as regards the design of the isolating valve D^1. The special object of the arrangement of the isolating valve D^1 is to keep it in position under normal working conditions, quite independent of the position of the stop valve or any movement due to the rolling action of the vessels. For this purpose valve D^1 is provided with a small valve at L, which rests in a seat formed at the top of the hollow guide spindle M, which opens to the atmosphere at N. The valve is thus kept in position by direct pressure of steam. In case of accident the increased velocity of steam, consequent upon the difference of pressure, easily overcomes the pressure required to keep the valve in position under normal conditions, and closes the valve D^1 on the seating R. The valve D^1 is kept by the steam pressure against its seating R, and can only be released by shutting the stop valve C^1, when steam passing through the small openings P and S causes an equal pressure on both sides of the valve. The valve D^1 will then return to its seating L, and resume its normal position. When the valve D^1 closes, due to the occurrence of an accident, this is at once indicated by a small steam jet discharged from the opening N into the atmosphere. By adjusting the size of the opening N the isolating valve D^1 may be set to close at any difference in pressure. The valve C^1 acts in a similar way, and is in all respects the same as described in Fig. 88.

Ordinary double-beat valves are difficult to keep steam-tight, the two valves being rigidly cast in one piece; the two faces do not wear equally, hence while one may be steam-tight the other may not. Messrs. Holden & Brooke have introduced Green's valve, in which the difficulties met with in single and double-beat valves are overcome. There are two valves, as may be seen from the three sections in Figs. 90, 91, and 92. Fig. 93 shows the complete valve. It will be seen that the smaller valve L has the pressure on the under side, and the larger valve M the pressure on the top. By this arrangement the smaller valve naturally tends to open first on putting the operating gear into motion, and the smaller valve will continue opening until the pressure is equal on both sides of the larger valve; thus as soon as the larger valve is placed in equilibrium it can then, in its turn, also be opened with ease. The extent of lift on the lower valve is controlled by a stop on the cover.

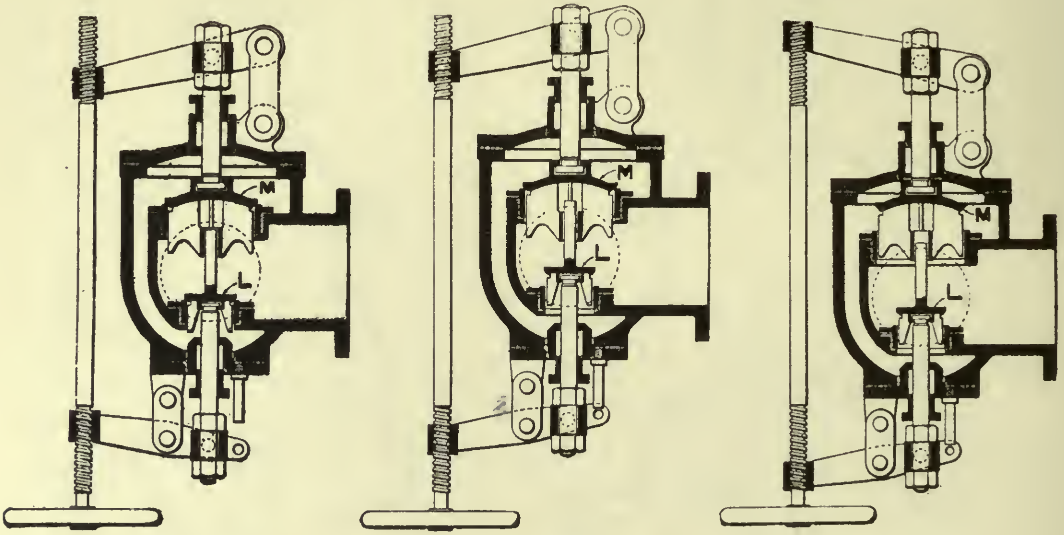


Fig. 90.—Shut.

Fig. 91.—Opening.

Fig. 92.—Open.

Green's Valve.

The valves and seatings are made of phosphor-bronze, unless otherwise specified, and the bodies are made in cast iron, steel, or gun-metal.

The valves are opened by revolving the hand wheel with a left-hand motion; the spindle is thereby rotated, and the nut on the left-hand thread travels along the spindle in the direction of the hand wheel.

The upper nut on the spindle is held stationary owing to the pressure being on the top side of the larger valve, and is screwed with a right-hand thread; this causes the spindle to travel through the upper nut in an upward direction, at the same time as the lower nut is also travelling along the spindle in the same direction. This action opens the smaller valve, and continues until the pin on the lower lever comes in contact with the stop on the cover.

The pressure is now equal on both sides of the larger valve, and as the lower nut cannot travel farther in an upward direction, owing to the stop, the continual revolving of the hand wheel causes the spindle to travel through the lower nut in the opposite direction; at the same time, the upper nut will also travel in an upward direction, this action opening the larger valve.

The small valve may be used independent of the larger. These valves do not chatter, can be quickly shut or opened, and nicely adjusted for small quantities of steam.

The independent action of the two valves is a great advantage.

In using common single-stop valves it is always better to arrange the inlet so that the steam tends to open the valve. The valve will then open as soon as the spindle moves. Whereas, if the steam shuts the valve, any slackness between the spindle and valve must be taken up before the valve moves. And the valve has to be forced open against the whole steam pressure. This slight precaution is often overlooked.

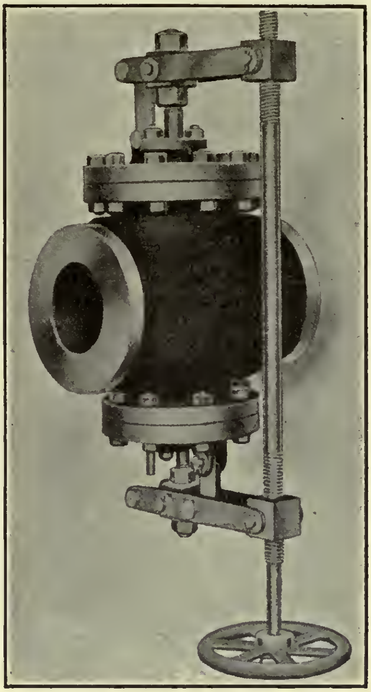


Fig. 93.—Green's Valve.

CONDENSERS

The foregoing engine parts are, generally speaking, common to all steam engines. Condensers, however, are only put on engines situated where water is to be found cheap and plentiful. The Modern Water Cooler plant is sometimes used where water is scarce. In this system the same water is used over and over again, with the addition of a small quantity of fresh water from town supply to make up for that lost by leakage, evaporation, and so on.

It is a moot point whether this system really effects any economy in the end. If we calculate the power required for pumps and fans, interest and depreciation, repairs, etc., on the cooling plant, cost of water lost, and put this against the horse-power gained by the vacuum, the margin of gain is not much in any case; and is nil in some cases. It is therefore a great point in laying down large steam powers to select, if possible, a site in reach of plenty of water. This obvious necessity for cheap water was in most of the early electricity generating stations entirely neglected. In Glasgow, with a river and canal available for water and carriage of coal and ashes, the generating station was planted far away from both river and canal,—water had to be paid for, and everything carted through the city streets. This is an example of electrical engineering to be found all over the country.

The blunders in the case of Glasgow have been rectified in the new stations erected since then.

Condensers may be divided into two classes. First, those in which the condensing water and condensed water mingle and run off together, in which case the pure condensed water from the steam is lost. Next, those in which the condensing water and condensed water are kept separate, so that the condensed water can be used for boiler feeding.

Oil in the engine cylinder is carried out by the exhaust steam and is a nuisance in the condenser, and a danger if it is carried in by the condensed water pumped into the boiler. Hence very little oil is used in engines like marine engines, in which the condensed water is used for boiler feed. A little oil is used at the start of a voyage, but none during the voyage, trusting to the cylinder condensation to lubricate the piston rings.

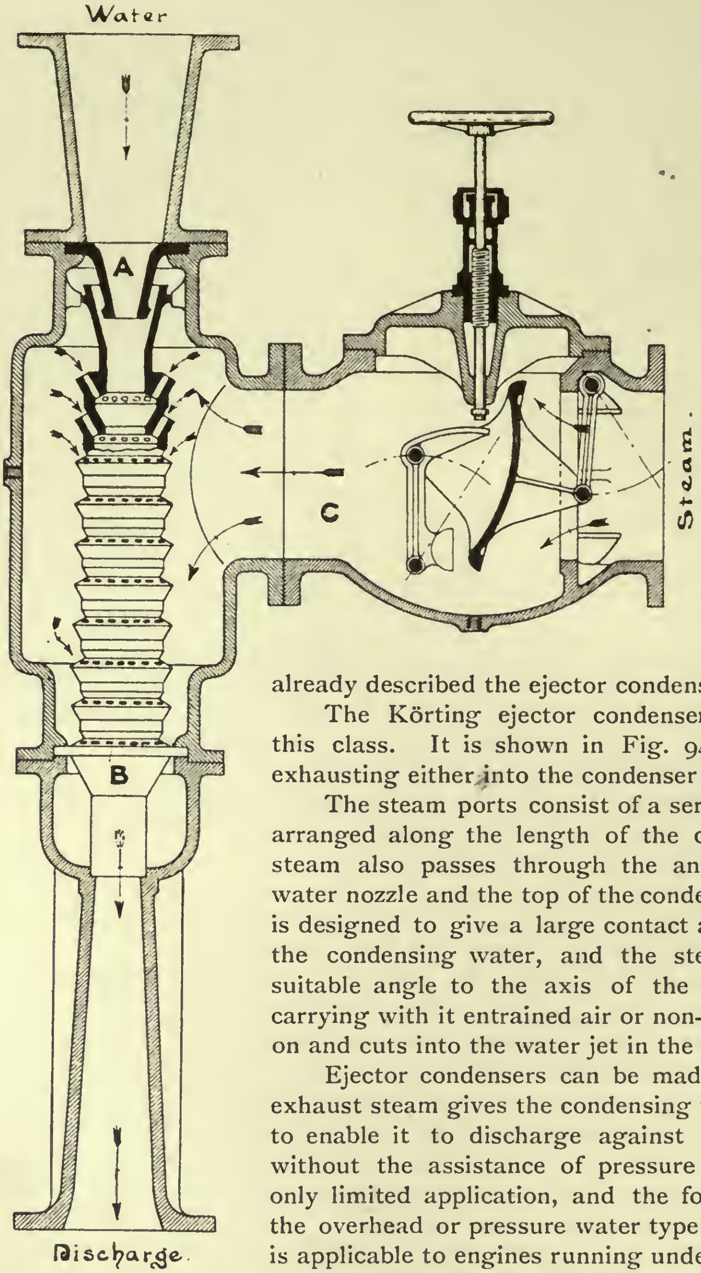


FIG. 94.—Körting Ejector Condenser.

Oil separators have been introduced in considerable numbers, but they are only partially successful. It is almost impossible to separate oil from steam or hot water, but if allowed to stand still and settle no doubt in time the oil and water separate by gravity. The best separator where stillness and time are not given is a cream separator working on the centrifugal principle, but it is doubtful if this principle is applicable to steam condensers.

In Volume I. we have already described the ejector condenser of Morton's invention.

The Körting ejector condenser is a very fine example of this class. It is shown in Fig. 94 with the 2-way valve for exhausting either into the condenser or atmosphere.

The steam ports consist of a series of obliquely drilled holes arranged along the length of the condensing tube, and some steam also passes through the annular opening between the water nozzle and the top of the condensing tube. The apparatus is designed to give a large contact area between the steam and the condensing water, and the steam ports are drilled at a suitable angle to the axis of the water, so that the steam, carrying with it entrained air or non-condensible gases, impinges on and cuts into the water jet in the direction of the flow.

Ejector condensers can be made so that the energy in the exhaust steam gives the condensing water the necessary velocity to enable it to discharge against the atmospheric resistance without the assistance of pressure supply water. They have only limited application, and the following notes refer only to the overhead or pressure water type of ejector condenser which is applicable to engines running under all conditions of variation.

Again referring to the illustration of the condenser, if water under a pressure equal to 15 feet water column, or say 7 lbs. per square inch, be supplied at the inlet nozzle, it will discharge from this nozzle with the velocity due to this pressure head. The water passing through the condensing tube B draws by inductive action, and carries with it part of the surrounding air, and a partial vacuum is thus formed in the condensing tube and the exhaust chamber of the condenser. The velocity and consequent momentum of the mixture of water and air passing through the cylinder at the bottom of the tube B is sufficient to overcome the atmospheric resistance, and the exhaust chamber is thus sealed on the discharge side. Assuming the engine to be standing, and all connections to the condenser to be air-tight, a nearly absolute vacuum is produced by the inductive action of the water jet. The absolute pressure at the nozzle A is due to the head of water plus the pressure of the atmosphere, or, say, $15+34=49$ feet water column, and as when first starting the condenser the exhaust chamber is under atmospheric pressure, the resistance to the discharge from the water nozzle A is equal to 34 feet head of water, and the water is delivered into the condensing tube at the velocity due to $49-34=15$ feet head. As the air is exhausted the counter-pressure diminishes, so that when the vacuum is equal to 30 feet of water column the velocity of the water passing into the condensing tube is that due to $49-(34-30)=45$ feet head. The water therefore passes through the condensing tube with gradually increasing velocity and momentum as the vacuum rises, and given the assistance of 15 feet head of injection water the class B overhead type condenser works with absolute certainty under any conditions of load variation, and the air and non-condensible gases are discharged with the water without the assistance of air pumps.

As to economy, the makers claim for the Körting condenser according to the following calculations:—

For comparison, a condensing plant for a 1000 brake horse-power engine may be taken. Assuming a steam consumption of 22 lbs. per brake horse-power, the plant would condense 22,000 lbs. of steam per hour, and a No. 35 Körting condenser, using 55,000 gallons of water per hour, would maintain a vacuum of 24 inches Hg. The condenser is 8 feet long, and with 15 feet head of water and a 2 feet long discharge pipe, the total difference between the supply and the discharge levels would be 25 feet, and the actual horse-power required would be $(550{,}000\times 25)\div(33{,}000\times 60)=7$. An efficiency of 60 per cent. can be obtained with electrically driven centrifugal pumps with full allowance for motor pump and dynamo losses. The actual brake horse-power required for working such a plant would be $(7\times 100)\div 40=17\frac{1}{2}$ brake horse-power, or less than 2 per cent. of the power developed by the main engine. In this calculation no allowance has been made for loss by friction in pipes or for gravitation flow from the hot well. This allowance has been purposely omitted, as it will vary in every case, and similar allowance would have to be made with any type of condensing plant.

It is often assumed that ejector condensers require a comparatively large quantity of water, but this is incorrect. With injection water at a temperature of 60° Fahr. and barometer at 30 inches, ejector condensers will maintain a vacuum of 24 inches Hg column with a proportion of water to steam of 25 : 1. In most instances the quantity of water used is of importance only in relation to the power required for working the plant, and in this respect an installation of ejector condensers compares favourably with either surface or jet condensing plant.

JET CONDENSER

So called because the incoming steam is met by a jet of cold water broken up into a spray. It instantly condenses the exhaust steam. The injected water and condensed water, together with air and oil, are pumped out by an air pump, and thus the vacuum is maintained.

A good design of this class of condenser is shown in Fig. 95 in section. The steam enters by the upper branch pipe, and meets the cold injection water sprayed up through a perforated injection pipe seen in the cross section. The water collects at the bottom. The pump has a horizontal barrel, with ports opening into the condenser through which the water, air, etc., fall into the pump barrel when the piston uncovers the ports at the end of the out-stroke. The fluids are then pushed out through the outlet valves at the other end of the pump barrel. The pump has no suction valves (which, of course, avoids the expense of renewal), no bypass valves or snifting cocks, and there are only the head valves to attend to, which are accessible through the end door. There is no bucket valve or packing to same. An increase of 1 lb. in the vacuum is generally obtained, consequent on the exceedingly reduced clearance of the air pump.

In working, the piston forms a vacuum in the barrel as it moves from the head valves, until it passes the inclined ports where the water from the injection and condensed steam and air rush in, the same action of the piston backing up the water on the other side and helping to force it through the ports; on the return stroke these ports close, shutting off the condensing chamber, and the water and air are then driven through the head valves. The valve on the top is for the inlet of air, so that the overflow can easily get away.

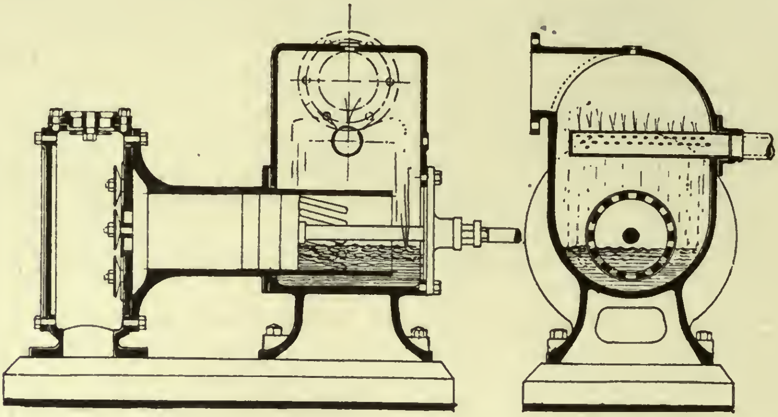


FIG. 95.—Jet Condenser and Air Pump.

The air pump barrel works perfectly without a bush, effecting a further economy in construction.

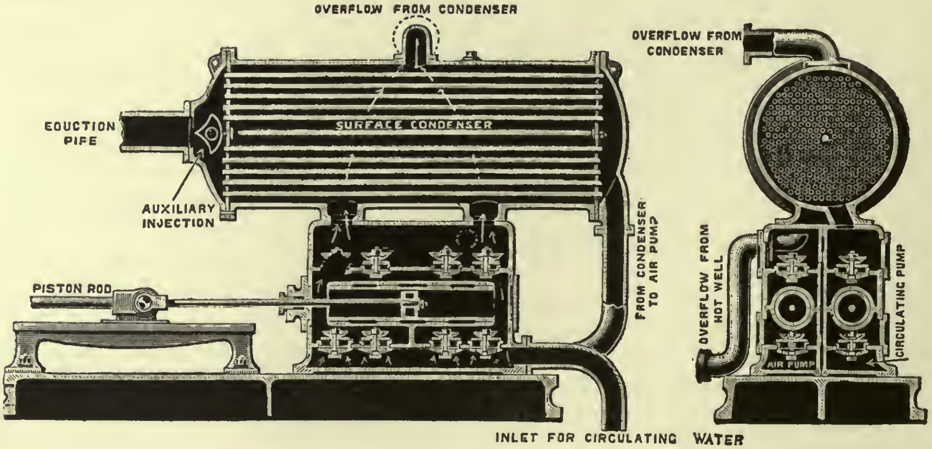

OVERFLOW FROM CONDENSER
OVERFLOW FROM CONDENSER
EDUCTION PIPE
SURFACE CONDENSER
AUXILIARY INJECTION
PISTON ROD
FROM CONDENSER TO AIR PUMP
OVERFLOW FROM HOT WELL
CIRCULATING PUMP
AIR PUMP
INLET FOR CIRCULATING WATER

FIG. 96.—Surface Condenser.

The hot well for the feed pump suction is arranged on a branch of the overflow pipe, which should in no case deliver against a "head," *i.e.* the water should flow away downwards by its gravity.

In both the foregoing types of condenser the waters are mixed. In order to keep them separate the surface condenser (Fig. 96) is employed. In this condenser the steam passes through a great number of the brass tubes, about $\frac{3}{4}$ inch diameter, and the tubes are kept cool by circulating the cold water around their interior by means of a circulating pump, generally a centrifugal pump. This condenser is necessary for condensing engines.

When the water supply is deleterious in character to an engine or boiler, as sea or muddy water, for instance, a surface condenser should be used, in which the steam is cooled without mingling with the condensing water; the illustration above shows one of the arrangements. The steam entering by the eduction pipe passes through the tubes and drains as condensed water into the air pump; the circulating pump is alongside, and drives the cooling water between the tubes and out at the top of the condenser. Both pumps are driven from a cross-head fixed to an extension of the engine piston rod, which, according to the size of the engine, is or is not fitted with under guide.

The condensed water is returned to the boiler, and is pure; no deposit can therefore be formed, and this alone effects a great saving in wear and tear of the boiler, and also in fuel.

From 50 to 80 gallons of circulating water per indicated horse-power per hour are necessary.

Both air and circulating pumps are bushed with brass, and the valve seats and fittings, rods, buckets, etc., are of brass, as also are the tube sheets and tubes, the latter being packed with screw glands and cotton packing, which is without question the best arrangement, and is adopted by the Admiralty universally.

The smaller sizes of condensers have a self-acting air and circulating pump, arranged at either end of the same pump barrel.

Both these examples of condensers are the products of Messrs. Ruston, Proctor, & Co., Lincoln.

KÖRTING WATER COOLER

The Körting water cooler (see full page Plate) acts upon the principle of evaporation. It is a principle used for cooling water in India and other hot countries. If water is kept in a porous pot and stood in a current of air, and shielded from the sun, the water which oozes out is evaporated. The heat required to evaporate it is taken from the water itself, and so the temperature is lowered. 1 lb. of water evaporating takes about 1000Θ, so that if 21 lbs. of water at 120° Fahr. is evaporated down to 20 lbs., the remaining 20 lbs. having furnished the heat, its temperature will be lowered by $\frac{1000}{20} = 50^\circ$ Fahr. $= 120 - 50 = 70^\circ$ Fahr. Where sufficient space is available, cooling by spraying the water into the air, so that each particle evaporates about one-twentieth of its weight, is a satisfactory method. 4 square feet of superficial surface of a pond or reservoir is required for every 20 lbs. of steam, or 1 brake horse-power per hour.

If less area is available some means must be adopted to draw off or blow off the saturated air to make room for fresh air.

With the Körting system of spray cooling, distributing pipes are carried over a pond or shallow spray collecting area, the distributing pipes being served by a main pipe (from which they are usually valved off) into which the water to be cooled is delivered under pressure. The distributing pipes carry at suitable intervals gun-metal nozzles of the construction shown in Fig. 97. These nozzles are fitted with threaded spindles, and the water travelling round the threads is given a rotatory movement, and thus, by centrifugal action, is divided into fine drops on issuing from the nozzles. This division of the water exposes a large surface to contact with the air, and the consequent absorption and evaporation of a portion of the water sprayed lowers the temperature of the remainder, which falls into the pond cooled for re-use. The jets of water issuing from the nozzles have an inductive effect on the surrounding air, and thus assist in drawing fresh supplies of dry air, and in discharging the saturated air above the level of the spray.

The power required for working the plant is small, as a pressure equal to 15 feet head of water suffices for the production of an efficient cooling spray. Taking a 1000 brake horse-power plant using 20,000 lbs. of steam per hour, about 60,000 gallons of water per hour, or say 1000 gallons per minute, would be required for condensing. With allowance for loss by friction in pipes, 20 feet head of water would be needed for the cooling plant, so that the theoretical horse-power required for this would be $(10,000 \times 20) \div 33,000 = 6$. If a centrifugal pump direct driven by an electric motor be used for giving the necessary pressure, a total efficiency of about 40 per cent. may be taken after allowance for dynamo, motor, and pump losses, so that the actual brake horse-power absorbed would be $(6 \times 100) \div 40 = 15$, or $1\frac{1}{2}$ per cent. of the total power developed by the engine. An installation of ejector condensers with spray-cooling plant would take for working less than 4 per cent. of the power developed by the main engines. The cooling will vary with the humidity and temperature of the atmosphere and the strength and direction of the wind, but with water leaving the condensers at 120° Fahr. the average temperature of the re-cooled water may safely be taken at 75° Fahr.

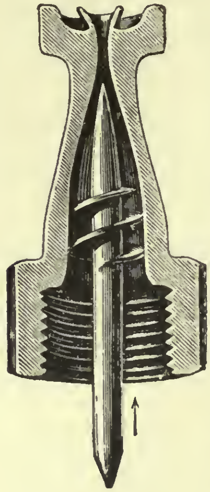


FIG. 97. — Körting Water - Cooling Nozzle.

The cooling pond need only be shallow, as for each foot depth there is sufficient water to keep the steam plant running for twelve hours should there be any interruption to the water supply. With a comparatively large body of water there is the advantage that there is not so great a rise of temperature and consequent drop of vacuum on the peak of the load, with unfavourable cooling conditions, as is the case with evaporative condensers or cooling towers, where the total quantity of water in use is relatively small.

A number of spray-cooling plants are at work in combination with ejector, jet, and surface condensers. The system has been adopted at the Cardiff, Cheltenham, Epsom, Glasgow, Kensington and Notting Hill, Wallsend, and Willesden Electricity Stations, and at a number of iron and steel works in the Cleveland and other districts.

A cheap and effective evaporative spray cooler is used in some Yorkshire mills. It consists of a wooden erection about 20 to 25 feet high, with 4 to 5 platforms, on which is laid a layer of brushwood. The water from the hot well is pumped up by centrifugal, and scattered over the top layer; it then drips and trickles down through the structure in tiny streams and drops, and is thus freely exposed to the air. By the time it reaches the pond at the bottom over which the structure is erected the water falls to a temperature about that of the atmosphere.

The effective horse-power of an engine is less than the indicated horse-power by the engine friction, but the indicated horse-power is itself less than the work done by the amount of back pressure. In condensing engines the back pressure is only 2 or 3 lbs. per square inch, whereas in non-condensing engines it is over 15 lbs. per square inch. Constant speed engines with a variable load, work against pressures which are nearly independent of the load.

Take a diagram (Fig. 98), in which the line *a*, *b* is absolute vacuum, and the cross shaded part the actual indicator card. The absolute indicated work is represented by the area *a*, *b*, *c*, *d*, *e*. The work wasted by back pressure is shown by area *a*, *b*, *c*, *f*, *e*. This area becomes a larger and larger percentage of the absolute work as the load diminishes.

Hence the reduction of back pressure is of most importance in variable load engines.

At full load a non-condensing well-designed engine takes very little more steam per brake horse-power than a condensing engine,—the back pressure of the non-condensing engine being balanced somewhat by the greater cylinder and port condensation in the condensing engine.

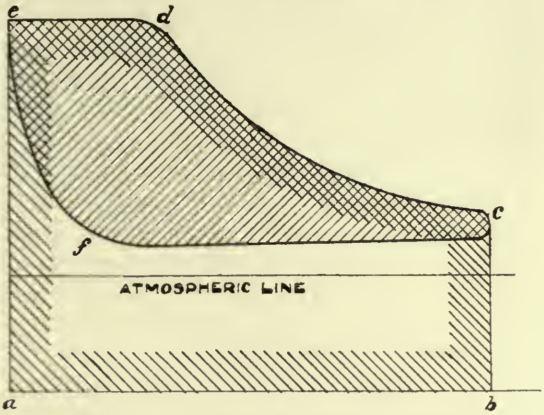


FIG. 98.—Condensing Engine Diagram.

In an electricity station, where variable load is the rule, condensing is therefore of importance, and warrants the outlay on water-cooling systems where a natural supply is not to be found. In one case the addition of condensers saved 50 per cent. of steam on variably loaded engines.

THE SYPHON CENTRAL CONDENSER

This type of condenser has some advantages in large installations. Fig. 99 shows the arrangement of the Worthington central condenser. This condenser is practically the same as that shown in sectional view in Fig. 100. The large pipe is the exhaust, and the left-hand pipe the injection water.

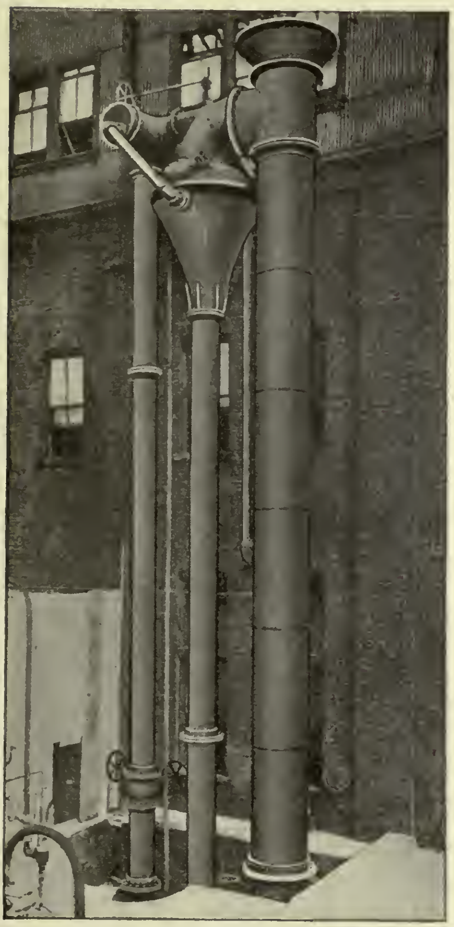


FIG. 99.—Worthington Central Syphon Condenser.

The Worthington system of central condensation, by which the exhaust steam of the engines of an entire plant may be condensed by one condenser, possesses features of value, and is being extensively introduced into large power stations, rolling mills, and blast furnace engine houses. It reduces the apparatus to the simplest form, and at the same time allows the use of more economical types of condenser and pumps than is possible with relatively small individual condensers for each engine unit. Attendance and wear and tear are reduced to a minimum, and the degree of vacuum obtained is higher than that with ordinary air pumps and condensers. Any number of engines scattered over an extended area may be exhausted into the system, and part of them may be started up and shut down without interfering with the others.

A characteristic of the system is the use of separate water and dry vacuum pumps. The former handles simply the water for condensation, which it lifts by suction, and discharges into the condenser. The condenser is placed in an elevated position about 30 feet above the hot well, and the water falls out of it by gravity against the pressure of the atmosphere. As the condenser has no connection with the suction pipe of the water pump, except through that pump, it will be seen that a fluctuation or sudden fall of the vacuum in the condenser does not cause a loss of suction as would be the case with the ordinary type of jet condenser, in which the vacuum in the condenser lifts the injection water. This feature makes the central condenser especially adapted to situations where there is apt to be a sudden overload thrown on the condenser, such as obtains in electric railway power houses and rolling mills. The water pump, as it has a uniform duty to perform, is usually provided with compound or triple-expansion steam cylinders. This places it on the same basis as regards economy as pumps employed in water works pumping stations.

As the water pump is assisted in lifting the water by the vacuum in the condenser it is readily primed on starting up, and the load upon the pump reduced in proportion to this benefit on regular work. The former feature is of value when long suction pipes are used. The vacuum pump may be started up and the air removed from the suction pipe ; the water will then flow in until above the pump valves, when the pump can be immediately put into service.

The duty of the vacuum pump is to exhaust the air from the condenser and the pipe system. This pump is designed on the most economical lines, which is made possible by the fact that the air is cooled by an air cooler in the injection pipe, and is practically free from water.

It may, however, be pointed out that while it is in many cases sufficient to draw all the exhaust steam from scattered engines to a central condenser, in order to save water and reduce back pressure on the engines, a very high vacuum cannot be obtained in the cylinder of an engine if the exhaust pipes are long. For the velocity of the steam is great, and friction retards it. Still, in ordinary cases the vacuum in the cylinders may be 23 to 24 inches mercury.

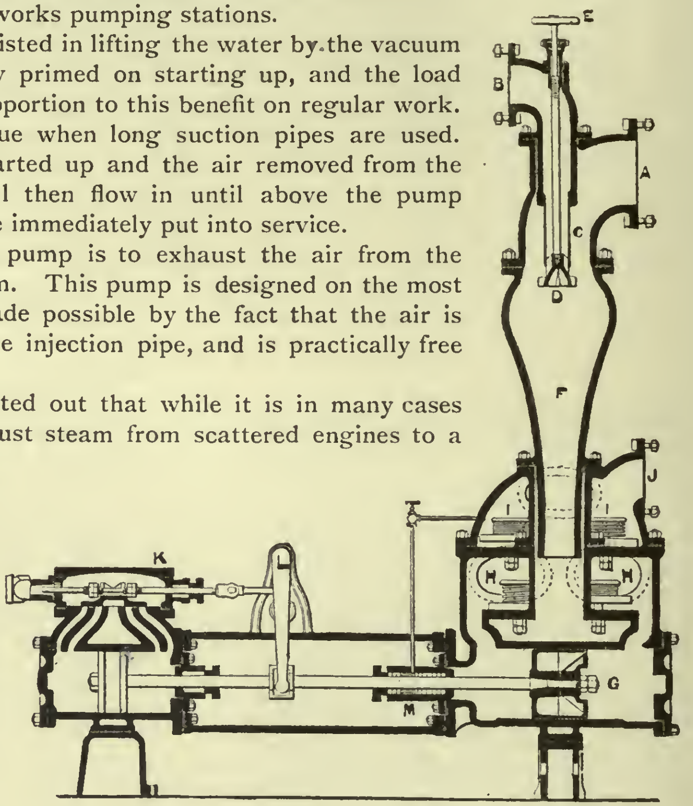


FIG. 100.—Worthington Jet Condenser and Air Pump.

The Worthington jet condenser and pump are shown in section in Fig. 100. It is designed with an eye to utilising the velocity and momentum of the water in assisting to move the pump. The injection water enters by B, and is regulated by hand wheel E. The exhaust enters by A, and the condensation takes place in F.

Exhaust steam from an engine enters a vacuum with a velocity of about 1900 feet per second, and water under atmospheric pressure with a velocity of 47 feet per second. Excepting the machine herein described, it may be said that in all forms of jet condensers operated by air pumps the injection water and the water from the steam fall to the bottom of the condensing chamber, come to a rest before entering the pump, and consequently lose the value of the momentum acquired when entering the vacuous space.

The air set free by the condensation of the steam is intermingled with the water. In fact, it acts as a water pump, although the water is aerated. There is here a great distinction between pumping aerated water, and an air pump pumping air and water unmixed. In the latter case the air pump has a varying and irregular duty to perform, and the inevitable result is an irregular and slamming movement.

The idea is to maintain the velocity of the water by delivering it into the pump through the converging nozzle, so that its velocity becomes pressure behind the pump piston, and the entrained air is carried along with the rapidly moving water into the pump, no time being allowed for its disentanglement.

THE COOLING-TOWER SYSTEM

The Worthington cooling-tower system (Fig. 101) may be taken as typical. The system is to be recommended only where space is not available for evaporative cooling, and where engines can take the greatest benefit from the vacuum.

The water from the pump, at a temperature of 120° Fahr. or thereabout, is carried to the top of a tower as shown in Fig. 101. And this tower contains tiers of tiles or drain pipes, down through which the water falls in thin films and drops. A fan meanwhile maintains an upward blast of air. By this means rapid evaporation occurs, and the vapour carries off heat, so cooling the water.

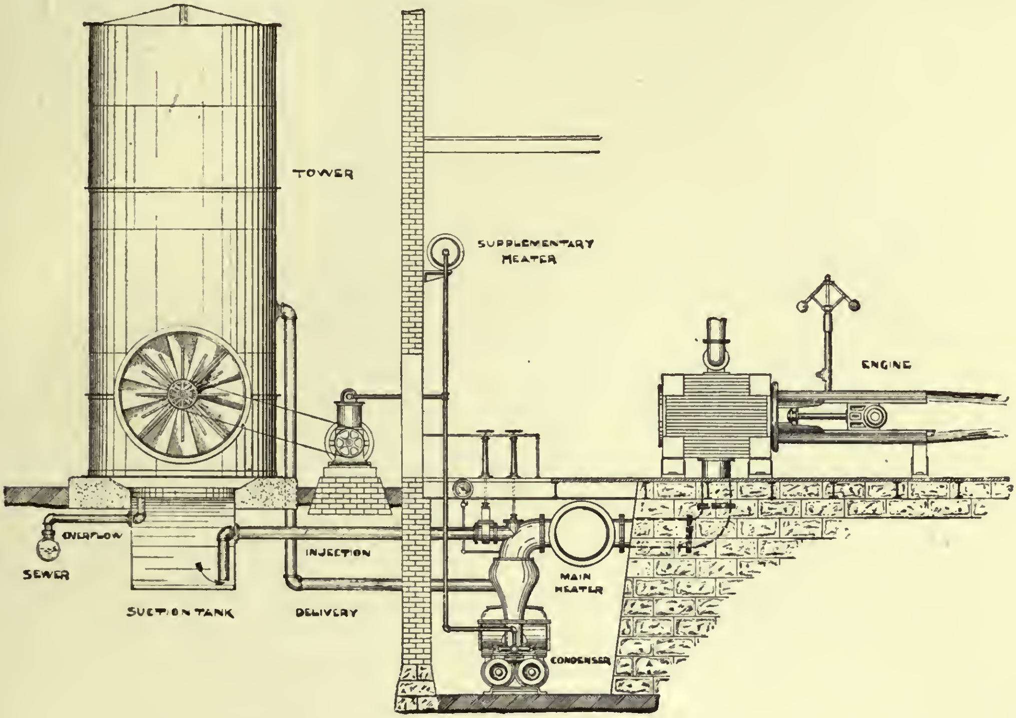


FIG. 101.—Worthington Cooling-Tower System.

Many different arrangements of this plant are in use, but the illustration here given shows the general plan.

FEED PUMPS

Intimately connected with the question of condensers is the boiler feed pump question. The Worthington feed pump so much used is shown in Fig. 102 in section, and Fig. 103 a complete view.

The valve motion is the prominent and distinguishing peculiarity of the Worthington pump. To this it owes its complete exemption from noise or concussive action. Two steam pumps are placed side by side, and so combined as to act reciprocally upon the steam valves of each other. The one piston acts to give steam to the other, after which it finishes its own stroke, and waits for its valve to be acted upon before it can renew its motion. This pause allows all the water valves to seat quietly, and removes everything like harshness of motion.

The illustration is a sectional view of one side or half of a high-pressure steam pump. Its object is to exhibit the great simplicity of its interior arrangement, especially that pertaining to the steam valve.

This valve, as may be seen at E, is an ordinary slide valve, working upon a flat face over ports or openings. Its simplicity and durability, in contrast with any other form of steam valve, is well known. Although numerous attempts have been made to supersede it, it still maintains its place on locomotives and other forms of high-pressure crank engines. No matter how long the engine may stand inactive, a slide valve will not rust or adhere to its seat, and is always ready to start when required. No water can collect in its cavities to produce trouble by freezing. In a word, it may be called the simplest and most reliable steam valve known to engineers.

In the engine the motion of this valve is produced by a vibrating arm, seen at F, which swings through the whole length of the stroke with long and easy leverage. As the moving parts are always in contact, the blow inseparable from the tappet system is avoided. Even the motion of the well-known eccentric upon crank engines is not comparable to this for moderate friction and durability.

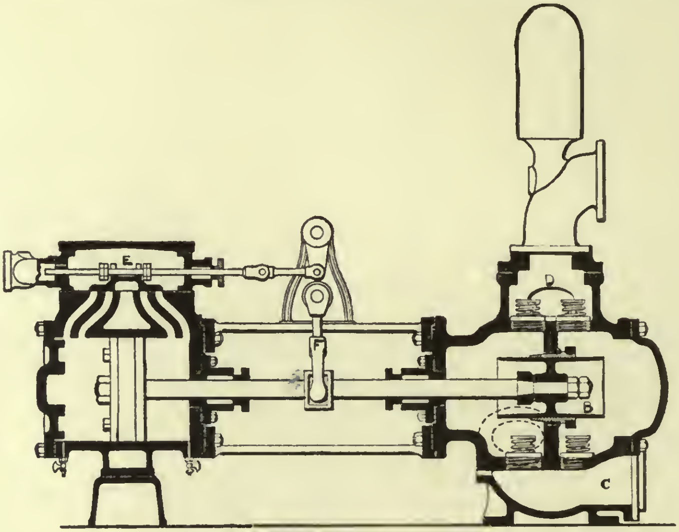


FIG. 102.—Worthington Pump. Section.

The arrangement of the double-acting plunger is shown at B. It works through a deep metallic packing ring, bored to an accurate fit, being neither elastic nor adjustable. Both the ring and the plunger can be quickly taken out, and either refitted or exchanged for new ones at small cost; and if it be desired at any time to change the proportions between the steam pistons and pumps, a plunger of somewhat larger size, or decreased to any smaller diameter, can be readily substituted. As exact proportions between the power and work are always desirable, if not necessary, this is a very important advantage.

This system of renewal of the working parts has been proved by long experience to be the least expensive and most satisfactory for ordinary work. The plunger is located some inches above the suction valves, to form a subsiding chamber, into which any foreign substance may fall below the wearing surfaces. This enables it to work longer without renewal than the usual form of piston pump, especially in water containing grit or other solid material. The water enters the pump from the suction chamber C, through the suction valves, then passes partly round and partly by the end of the plunger, through the force valves, nearly in a straight course, into the delivery chamber D, thus traversing a very direct and ample water way. The bottom and top plates furnish a large area for the accommodation of the valves. These consist of several small discs of rubber or other suitable material, easy to examine and inexpensive to replace.

The following directions and explanations will enable the engineer to adjust the pump and take up for wear:—

The steam valve has no outside lap, consequently, when in its central position, it just covers the steam ports leading to opposite ends of cylinder.

By *lost motion* is meant the distance a valve rod travels before moving the valve; or, if the steam chest cover is off, the amount of lost motion is shown by the distance the valve can be moved back and forth before coming in contact with the valve rod nut.

To set the piston in the middle of its stroke, open the drip cocks and move piston by prying on the cross-head (not on lever) until it comes in contact with the cylinder head; make a mark on piston rod at face of steam end stuffing box follower; move piston back to contact stroke at opposite end. Make second mark on piston rod half-way between first mark and the aforesaid follower. Then if piston is again moved until second mark coincides with the face of same follower, it will be exactly in the middle of its stroke.

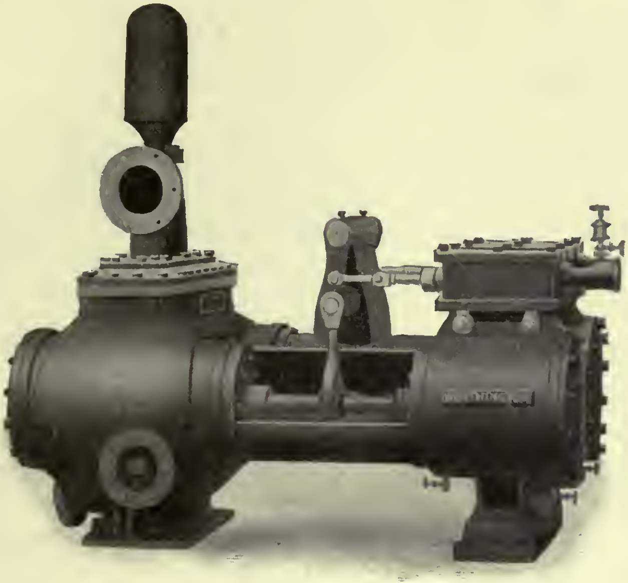


FIG. 103.—Worthington Pump.

Bear in mind that the piston on one side moves the valve on opposite side.

(*a*) When the steam valve is moved by a single valve rod nut. Place one piston in the middle of its stroke; disconnect link from head of valve rod on opposite side. Then set the valve in its central position; place valve nut evenly between jaws on back of valve, screw valve rod in or out until eye on valve rod head comes in line with eye of valve rod link; then reconnect. Repeat the operation on opposite side, and the valves will be properly set.

(*b*) When the valve rod has more than the lock nut. Place one piston in the middle of its stroke and the opposite slide valve in central position; adjust lock nuts so as to allow about $\frac{3}{16}$ inch lost motion on each side of jaw, and valve is set. Do not disconnect the valve motion. Repeat operation on other side.

The best way to divide the lost motion equally is to move the valve each way until it strikes the nut or nuts, and see if the port openings are equal.

It is advisable to place both pistons at the middle of their strokes before touching either slide valve.

Too much lost motion will tend to lengthen the stroke, and may cause the piston to strike the cylinder heads ; *vice versâ*, when there is not enough lost motion the stroke will be perceptibly shortened.

Another type of feed pump is shown in Fig. 104, by Thwaite Brothers of Bradford. It is single cylinder, whereas in the Worthington there are two cylinders and two pumps, the valves of the one cylinder being operated by the piston of the other. In the Bradford pump the valve gear will be clearly seen from the illustration, the special feature being the ring of holes at each end of the pump barrel. The distribution of steam in the actuating cylinder is governed by a single valve, which is operated entirely by mechanical means, and which is therefore not liable to stick in the annoying manner that is sometimes experienced with pumps of this nature fitted with an ordinary shuttle valve. The valve is operated on the "lost motion" principle by means of a lever connected to the piston rod, as shown in our illustration, the distribution of steam being indicated by means of arrows. There are bored in the liner of the pump cylinder a number of small holes or ports. These are so arranged that when the steam valve reaches its central position, and the steam is cut off from entering the steam cylinder, these ports are beginning to open, owing to the upward or downward movement of the piston pump. The effect is that the opening of the ports throws both sides of the pump piston into communication. The pressure on the pump piston is therefore equalised, and the expansion of the steam in the steam cylinder is then sufficient to complete the full stroke, thereby actuating the steam valve, so that steam is admitted to the opposite side of the steam cylinder. With this type of pump the steam is carried practically full stroke, no more work being done by expansion than is required to complete the very small portion of the stroke necessary to operate the valve. A full stroke is obtained all the time.

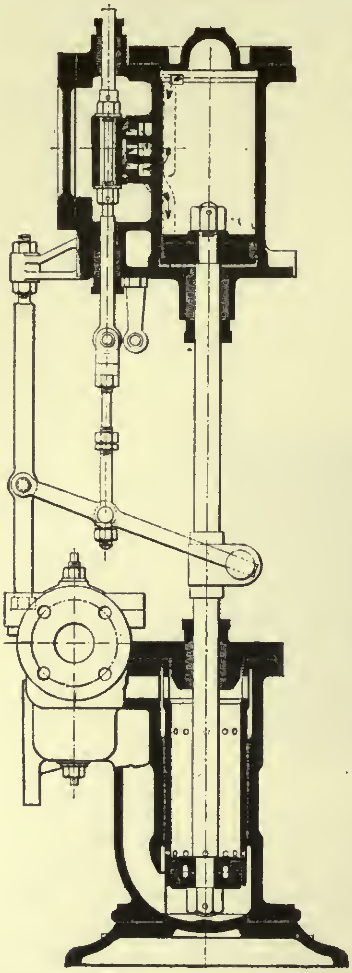


Fig. 104.—Thwaite Pump.

Several pumps exist in which the steam is used expansively,—a very desirable object when pumps are of large size. The Hessler pump is a good example of this class.

The special feature of the pump lies in the fact that, though it is without a fly-wheel, the steam is worked expansively. The general nature of the simple linkage which permits of this being done will be best understood by reference to Fig. 106. As there indicated, each piston rod is connected by a link to a crank or rocking bar, and these rocking bars are coupled together to a connecting rod. In Fig. 105 the cranks are represented by *d* and *e*, whilst *a* represents the coupling rod between them. The two pistons work one-half a stroke in advance of the other, and the effect of the linkage is such that as the pressure falls below the mean in one cylinder, the corresponding plunger receives power through the coupling rod from the other cylinder. The high-pressure valve is driven from the coupling rod *a* by the rocking sector shown at *g*. This sector is connected by the link *k* to the rocker *e*. The low-pressure valve is driven by coupling it to a second floating lever shown above *a*. One end of this lever is slotted to take the block *i*, which is mounted on the rocking lever *e*. The other is connected to the second rocking lever *d* by the pin *h*, which is adjustable so that the motion of the low-pressure slide valve is compounded of the motion of the two ends of the floating lever. Adjustments are also provided for the high-pressure slide valve, so that the cut-off in this cylinder can be varied between three-eighths and three-fourths of the full stroke.

Since the stroke of the plungers is fixed and constant, there is no trouble on starting the pumps, or when exhausting air from a long suction line even with a vacuum of 25 feet of water. Foot valves therefore are not fitted. The area through the suction valves is 170 per cent. of the plunger area, and the water passages have a correspondingly large section. The air vessel is, it will be seen, between the cylinders, and serves to replace framing, which would otherwise be necessary.

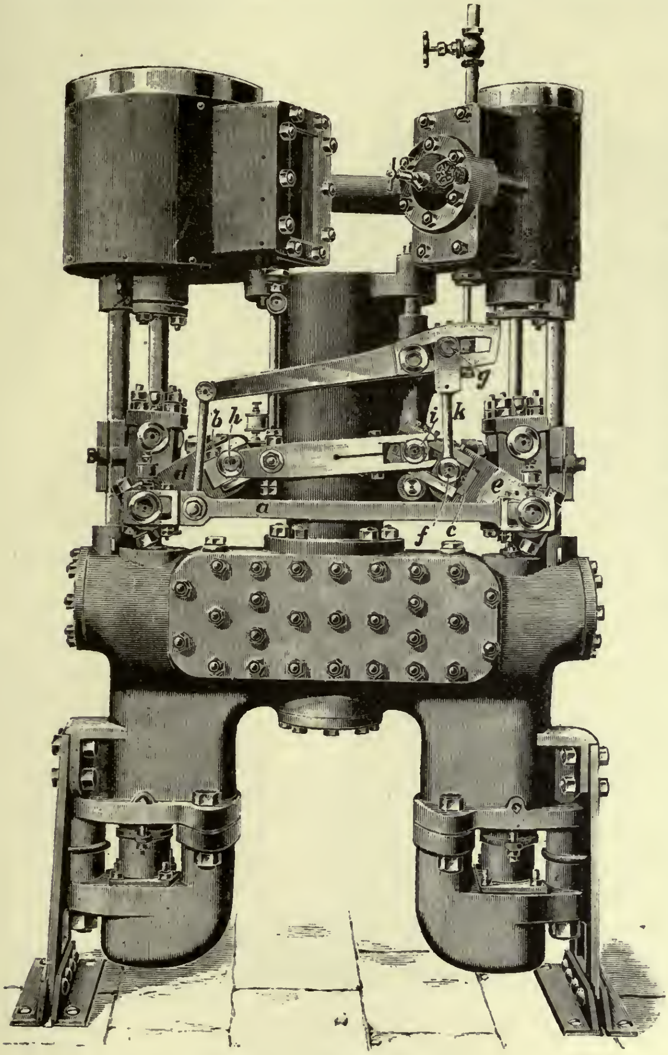


FIG. 105.—Hessler Pump.

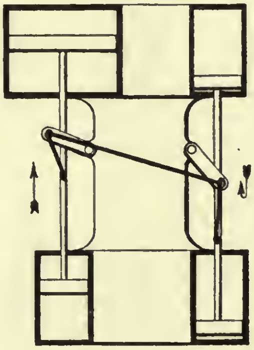


FIG. 106.—Diagram of Hessler Pump.

The best feed pump for most purposes is a steam injection, or exhaust steam injector. These have been already fully described in Volume I.

A feed water heater is a necessity. There are two kinds : the exhaust steam feed water heater, and the furnace gas feed water heater, commonly called the "Economiser."

It is doubtful practice to place a feed heater in the way of steam from the exhaust of an engine to a condenser. It lowers the vacuum and introduces troubles with condensing engines. The economiser is the best feed heater. But with non-condensing engines the exhaust steam should be used not only to feed the boilers through an injector, but also to heat the feed water to the highest possible degree. An example of a feed water heater by exhaust steam is shown in Fig. 107. It consists of a cylindrical shell with internal tubes. The feed water passes through the tubes on its way to the boiler, while the exhaust steam flows freely between the tubes.

The feed water has plenty of room in the lower chamber to come to rest and deposit any precipitates thrown down by heating, and these are blown off from time to time from the lower blow-off. Floating impurities collect as scum in an upper basin, and are blown off by the surface blow-off. This keeping clean of the interior of exhaust steam feed heaters is of great importance. Some of them have no blow-off or facilities for cleaning and in course of time choke up and throw great resistance against the feed pump. When a feed pump shows signs of hard labour it is in most cases due to choking up of pipes or feed water heaters.

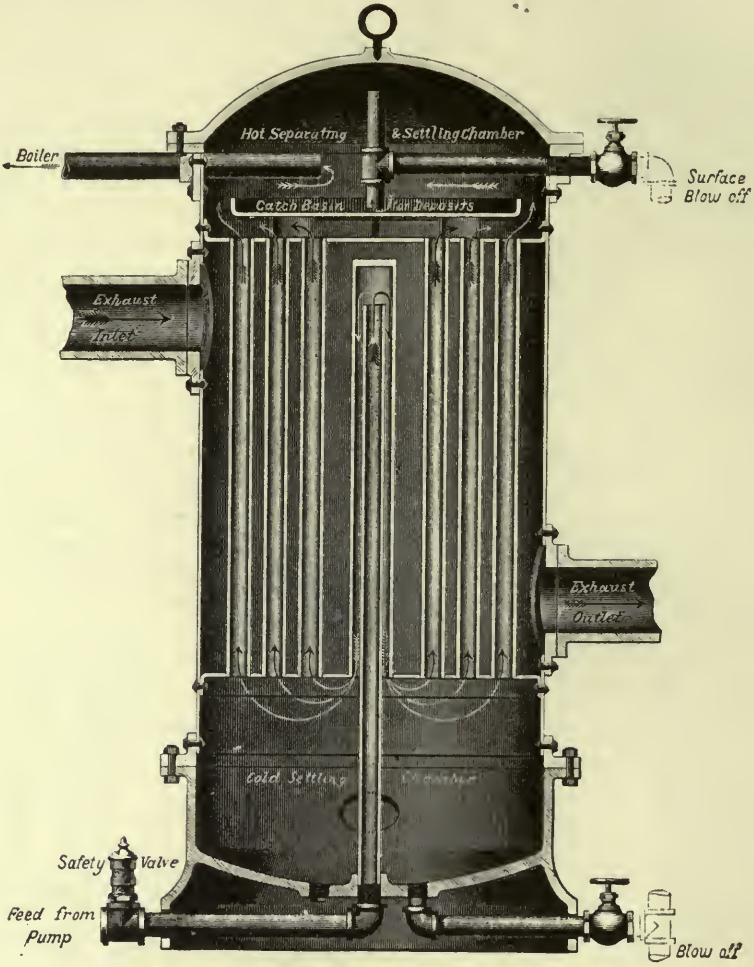

Hot Separating & Settling Chamber
Boiler
Surface Blow off
Catch Basin
Iron Deposits
Exhaust Inlet
Exhaust Outlet
Cold Settling Chamber
Safety Valve
Feed from Pump
Blow off

Fig. 107.—Feed Water Heater. Exhaust Steam.

The economiser shown in Figs. 108 and 109 is that constructed by Messrs. E. Green & Son, Wakefield, largely used for the purpose of feed heating. In this economiser great attention has been paid to the production of sound joints, perfect metallic joints between the pipes, and the lids which close their upper ends. They are built in sections, and are therefore readily erected and easily accessible for repair. By means of doors and a by-pass flue the economiser may be cut off at any time, and by means of smaller doors shown at the side of the tube chamber the hot gases can be partly deflected to moderate the heating.

Scrapers are provided, worked by power to slowly traverse the pipes and keep them free from soot, and the soot collects at the bottom in a soot chamber, from whence it is easily removed.

It is well to provide a bypass for the feed water direct to the boiler to be used in case of repairs to the economiser necessitating its stoppage, and the feed pipe from the pump to the economiser should be led through one of the boiler flues in order to take the chill off the cold water. If this is not practicable a small jet of hot water may be introduced.

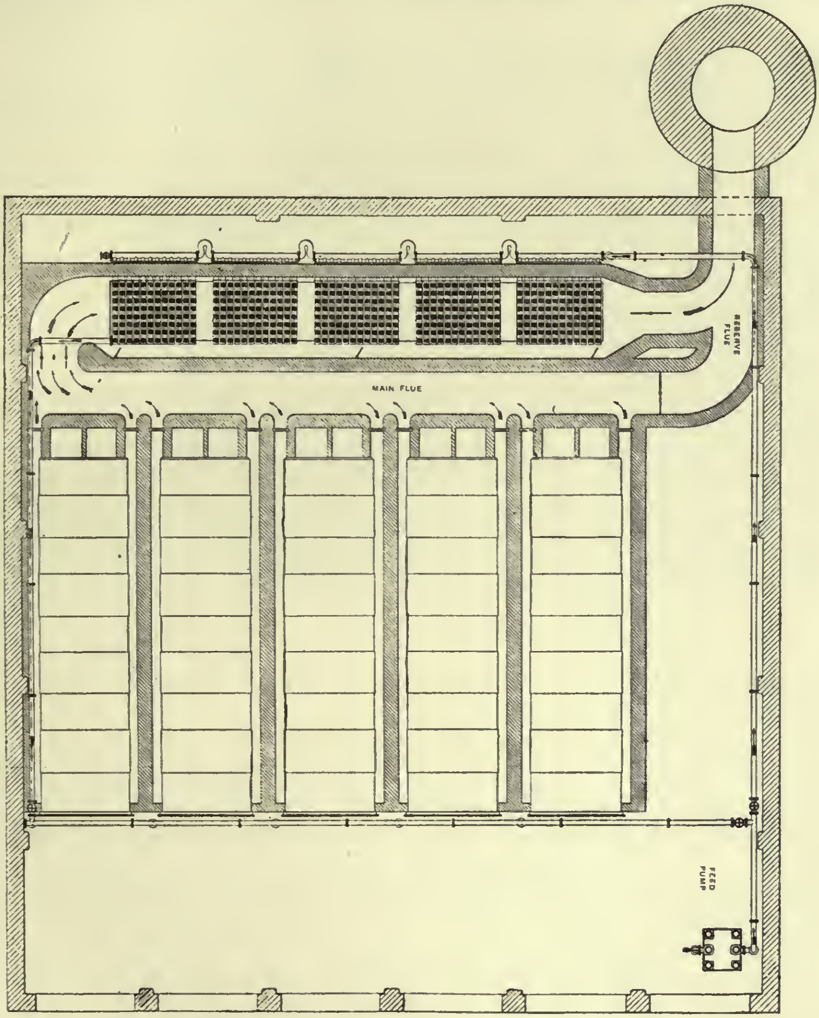


Fig. 108.—Green's Economiser. Feed Water Heater.

The economiser does not cost much at first, and little for upkeep afterwards, and the makers undertake regular inspection, which greatly relieves the owner of troubles which might arise from neglect.

Fig. 108 represents a common plan for laying down an economiser for a battery of four boilers. The main flue, it will be seen, can be opened to the chimney through the economiser or through the reserve flue direct to the chimney.

In Fig. 109 are shown the tubes and scrapers. The makers give the following explanations regarding the dimensions of an economiser for any given power or water evaporated. It is usual to allow four tubes for each ton of coal consumed per week. Thus, for example, if 20 tons of coal are consumed per week—of say 50 to 60 hours—the economiser should contain not less than 20 × 4 = 80 tubes. If the coal consumption is not readily ascertainable, another convenient rule is to calculate one tube for every 3 indicated horse-power. Thus, 300 indicated horse-power would require an economiser of about 100 tubes, 300 ÷ 3.

On account of the gearing arrangements the number of tubes in an economiser must be increased or diminished by not less than four sections. Thus if the economiser is four tubes in width its size requires to be altered 4 × 4 = 16 tubes at a time. If it is six tubes in width the alteration in size is 6 × 4 = 24 tubes at a time, and so on.

The heating surface of Green's economiser is calculated at 10 square feet per tube of 9 feet long, not including the top and bottom boxes. Thus an economiser of 144 tubes would have 1440 square feet.

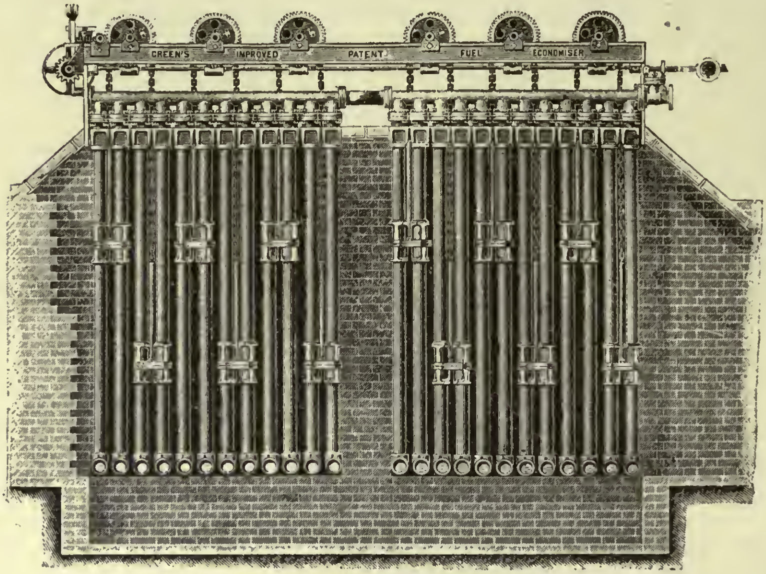


FIG. 109.—Green's Economiser.

The quantity of water in economiser when full is $6\frac{1}{4}$ gallons per tube; hence number of pipes × $6\frac{1}{4}$ = capacity in gallons.

Where boilers are being forced the saving effected is greater than in cases where boilers are easily worked. To ascertain the saving to be expected by any economiser at any boiler pressure, take the initial temperature of the feed water as in first column of the table below, and multiply the number of degrees the feed is heated by the decimal number in the column under working pressure. Example—boiler pressure, 160 lbs.; feed water temperature before heating, 80°; after heating, 260°; difference 180, which, multiplied by the figure in the table, 180 × .0872 = 15.7 per cent. saving.

Initial Temperature of Feed.	Working Pressure in Boiler (Lbs. per square inch).											Initial Temperature of Feed.
	0	20	40	60	80	100	120	140	160	180	200	
50	.0886	.0875	.0868	.0864	.0860	.0857	.0854	.0852	.0850	.0848	.0846	50
60	.0894	.0883	.0876	.0872	.0867	.0864	.0862	.0859	.0856	.0855	.0853	60
70	.0902	.0890	.0884	.0879	.0875	.0872	.0869	.0867	.0864	.0862	.0860	70
80	.0910	.0898	.0891	.0887	.0883	.0879	.0877	.0874	.0872	.0870	.0868	80
90	.0919	.0907	.0900	.0895	.0888	.0887	.0884	.0883	.0879	.0877	.0875	90
100	.0927	.0915	.0908	.0903	.0899	.0895	.0892	.0890	.0887	.0885	.0883	100
110	.0936	.0923	.0916	.0911	.0907	.0903	.0900	.0898	.0895	.0893	.0891	110
120	.0945	.0932	.0925	.0919	.0915	.0911	.0908	.0906	.0903	.0901	.0899	120
130	.0954	.0941	.0934	.0928	.0924	.0920	.0917	.0914	.0912	.0909	.0907	130
140	.0963	.0950	.0943	.0937	.0932	.0929	.0925	.0923	.0920	.0918	.0916	140

THE CENTRIFUGAL CIRCULATING AND AIR PUMP

This pump is largely used on ships for circulating condensing water. A modern example is shown in Fig. 110, fitted direct to a compound high-speed engine. The pump requires to supply a large volume of water at low pressure, just the work well suited to a centrifugal.

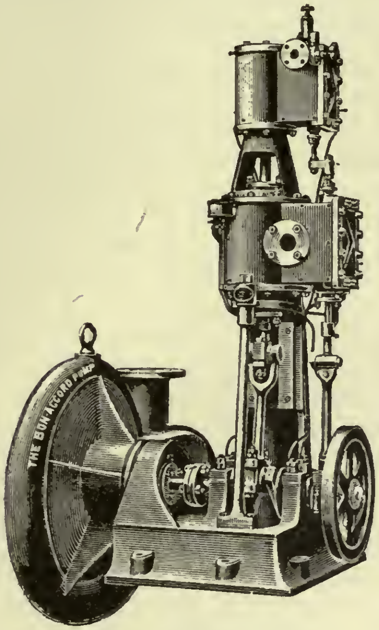


FIG. 110.—Circulating Pump and Compound Engine.

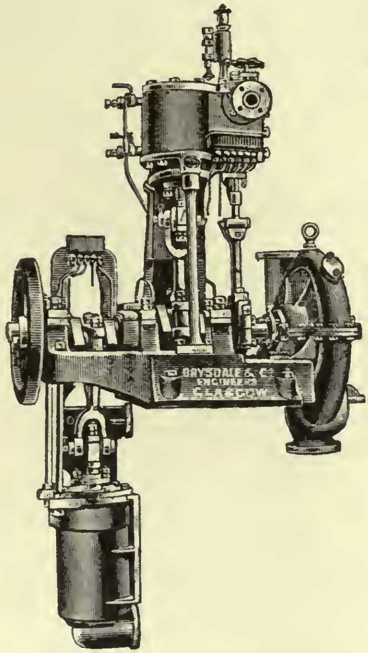


FIG. 111.—Circulating Pump, Air Pump, and Engine.

Fig. 111 shows the pump combined with an air pump, a convenient arrangement for turbines and other high-speed engines.

The preceding pages will convey the idea that the steam plant is somewhat complicated compared with the internal combustion engine plant. This is due to the highly developed state of steam engineering. The parts, though numerous, are in themselves simple devices, no very deep problems arising in their design.

The steam turbine shows more benefit from high vacuum than an ordinary steam engine, and there is no trouble about oil getting into the condensed water. The demand for high vacuum has resulted in several improved condensers and pumps.

One improvement of considerable interest consists in practically combining two condensers in series, so that the second one drains the last dregs of air and vapour from the first one.

For every additional inch of vacuum in a steam turbine above 25 inches a saving in steam equal to 6 or 7 per cent. is realised. This saving is due simply to the increased velocity and volume of the steam in the large buckets at the exhaust end of the turbine.

As a matter of fact, the low-pressure end of the steam turbine is the place where most of the work is done by the steam. This can be seen at once if a turbine is cut off from the condenser and opened to the atmosphere, and still more if the exhaust is throttled. Hence in some recent turbines the steam is rapidly expanded in two or three stages through Parsons type of wheel, and then through De Laval wheels in series into the vacuum.

If this necessity for a vacuum exists for all turbines it does not promise well for hot air or gas turbines, for in these prime movers the exhaust gases are not condensible, a chimney draught being all the relief from back pressure which can be provided.

WEIR'S FEED PUMPS

Steadiness of pressure and regularity of feed for modern boilers is one of the necessities of to-day. Messrs. Weir, of Cathcart, have met this demand in their pumps to be described.

The increasing adoption of water-tube boilers, with their attendant problems, has made a steady and reliable feed pump a necessity.

Further, as feed heating in some form or another is now general, only a pump of the slow-speed, direct-acting type is capable of dealing with water at the high temperatures commonly attained. Fly-wheel pumps, on account of the rise in pressures and their spasmodic working, are inadmissible for this purpose; while the dangerous strains to which they are liable also prevent their adoption for feeding at any of the pressures now common in central station practice. Professor Goodman, in a paper read before the Institution of Mechanical Engineers, describes some experiments on a plunger pump of the crank and fly-wheel type, in which the pressure in the discharge pipe rose to as high a point as 950 lbs. per square inch.

Besides higher economy secured by a single-acting pump compared with a duplex pump, this pump has been designed with special reference to its valves.

The steam valve of a direct-acting pump is one of its most important features as regards reliability and economy, and the Weir steam valve is acknowledged to be the simplest and best design yet produced for the attainment of these conditions. The perfection of working, certainty of action, steam economy, and freedom from complication obtained are distinctive features of the Weir arrangement, while the accuracy and superiority of the workmanship are unequalled in pump practice.

A brief description of the steam valve may be of interest. Referring to Fig. 112, it consists of a main and auxiliary valve. The main valve is for distributing steam to the cylinders; the auxiliary for distributing steam to work the main valve. The main valve moves horizontally from side to side, being driven by steam admitted and exhausted from each end alternately. The auxiliary valve is actuated by lever gear from the rod of the pump, and moves on a face on the back of the main valve, and in a direction at right angles to the main valve. By this arrangement there is no dead centre, the action being absolutely positive, because the only possible position in which the main valve can rest is at full travel—either for an up or down stroke of the piston.

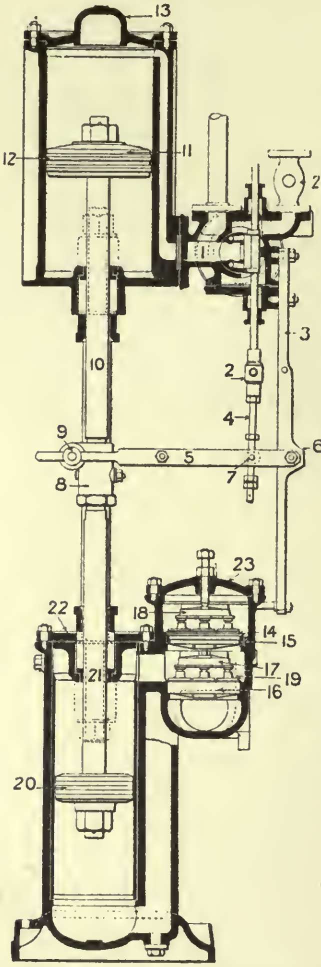


FIG. 112.—Weir's Feed Pump.

Both the main and auxiliary valves are simply slide valves, but the former is half round, the round side working on the cylinder port face, which is bored out on one side to fit the valve. On the back of this main valve a flat face is formed for the auxiliary valve to work upon. Both ends of the main valve are lengthened so as to project beyond the port face, and are turned cylindrical.

THE WEIR INDEPENDENT AIR PUMP

It is becoming more common to employ a separate air pump for condensers on engines instead of a combined air and water pump on the engine itself.

In stationary power plants the independent air pump has now been adopted for a considerable time, but the conditions on board ship have prevented the use of stationary land types. The tendency to employ separate engines for all auxiliary duties, noted above, and the increasing adoption of the marine steam turbine, where the effect of a high vacuum in increasing the efficiency is most marked, have attracted the attention of engineers to improved air pumps for high vacuum.

The independent air pump admits of a vacuum being formed and retained in the main condenser before the main engines are started, and at any time when they are stopped.

It enables the engines to be manipulated with greater ease and certainty at times when immediate response is of the utmost importance.

It relieves the engine-room of a considerable amount of complication, as the auxiliary condenser and auxiliary air pump, together with their piping, can be dispensed with.

Further, it enables the cylinders to be well and continuously drained, and avoids loss of water and unnecessary heating of the engine-room.

If the engines are stopped with heavy fires, it enables the steam to be blown into the condenser, and thus saves loss of water and blowing off at the escape pipe.

In conjunction with the steam turbine, however, the independent air pump is more particularly applicable, as air pumps cannot be satisfactorily driven from the turbine itself. There is the further consideration that a high vacuum is indispensable in securing the utmost efficiency from the steam turbine.

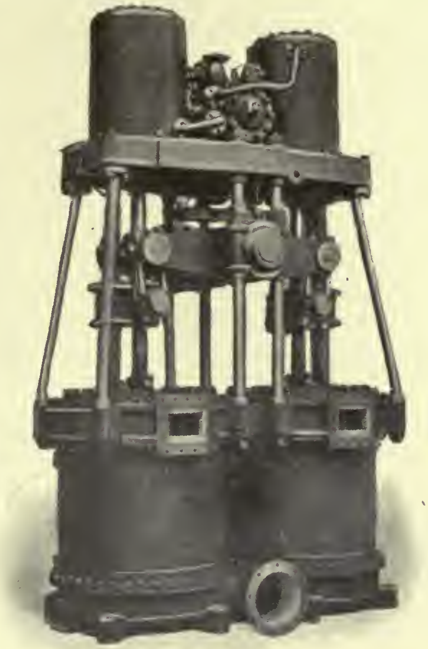


FIG. 113.—Weir's Independent Air Pump.

In view of the great increase in economy obtainable in turbine installations by securing a high vacuum, Messrs. Weir have designed and patented a special high vacuum plant for use in conjunction with these. The results of this design of plant in actual working have proved the high efficiency of the system with flat ends. Caps are fitted on each of these ends forming cylinders, which are closed at the mouths by the flat ends of the main valves, which act as pistons.

The function of the auxiliary valve is to admit steam through the ports on the back of the main valve to move the main valve from side to side. The ports for admitting steam to the top and bottom of the cylinder are arranged to cut off before the end of the stroke, and so slow down the pump, thus permitting the water valves to settle quietly and relieve the connections from any shock. On the last quarter of the stroke the steam is thus used expansively, so effecting a considerable economy in steam consumption. Provision is made, however, by turning round the caps covering the end of the main valve for admitting live steam during the entire stroke, as, when the pumps are starting and the metal is cold, the steam condenses and it is necessary to clear out the chambers of water. These caps are turned by means of the gun-metal spindles with indicating pointers at each side of the steam valve chest. When the pump is fairly started these bypasses—one for the up-stroke and one for the down

—are closed till the pump is working silently. The main and auxiliary valves are practically the only two moving parts in the valve chest. The stroke can be adjusted while the pump is working by the nuts on the valve spindle in accordance with the centre punch marks on the front stay, and is constant.

AUTOMATIC FEED REGULATOR

In many plants it is found to be of advantage to have the boiler feed regulated automatically, and for this purpose a float tank and regulating gear, as shown in Fig. 114, was designed. The tank or hot well is of cast iron, and can be placed in any convenient position relative to the pump. It contains a float pan with water-tight bottom and sides, and open on the top, suspended on two levers, so as to move up and down with a parallel motion. The top lever spindle is carried through the door at one end, and is balanced by a lever and weight. The float is always full of water, and the weight is adjusted to balance it when one-half is immersed in water. To the weight lever another lever is attached, which actuates the throttle valve controlling the supply of steam to the pump. The gear, of course, can be varied in form and arrangement, and fitted to existing tanks where necessary. A facing is left on the tank for attaching an air pipe, which should be led up as high as the source of supply. These cast-iron float tanks, however, are made in standard sizes, with regulating gear complete, and with throttle valve to suit the various sizes of pumps. An arrangement specially adapted for electric light stations using surface condensers, consists of two Weir feed pumps, with standard float tank and automatic control gear. Each pump works independently of the other, and draws from the float tank, which is connected to both. By means of the float regulation the speed of the pumps corresponds with the amount of water to be discharged. During the day, when the load is light, one pump is capable of doing the entire work, while towards night, when more boiler power is operated, the second pump is put on.

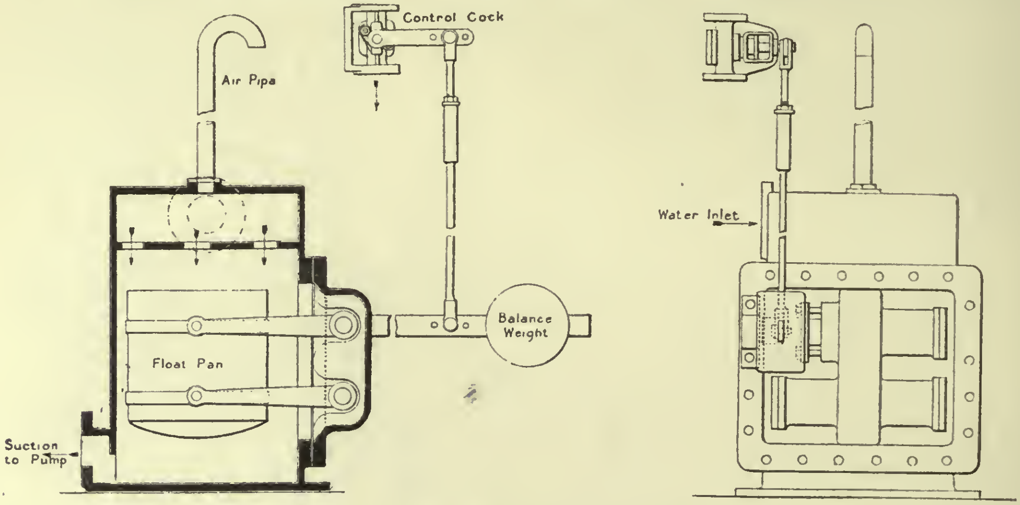


FIG. 114.—Weir Automatic Feed Regulator.

KINEMATICS OF STEAM ENGINE

The crank and connecting rod and the valve gearing come under problems to be solved by kinematics. The fly-wheel comes under kinetics. The reciprocating engine of necessity requires guides or links to convert the motion into a rotary motion. The old school of engineers following Watt delighted in beams and parallel motions (see Fig. 1 of this Volume), but soon it became evident to the mechanician that, however beautiful links and geometrical problems and their solution of parallel and other motions were, they were only at best necessary evils introduced for want of tools and machinery to make accurate slides. They all suffer from a multiplicity of pinned joints, and pinned joints, if properly made to transmit large stresses and at high-speed movements, are very expensive and require continual lubrication and attention (see Connecting Rod Ends). For the reason that pinned joints in engines are at best a source of trouble, they were reduced to only two—the two ends of the connecting rod. In the steam turbine they have been entirely abolished, and that is one reason for its rapid advancement.

In Watt's parallel motion there are no less than five joints; in Peaucellier's parallel motion there are seven links and six joints. As the day of links and joints is gone in steam engineering, we need not go into their construction here further than they concern the connecting rod.

Kinematic links and pins are all quite admissible in mechanism designed only to transmit and direct motions, such as the links in the motion of an engine indicator, but immediately they are subjected to the strains due to transmitting heavy pressures, as in engines, they introduce trouble.

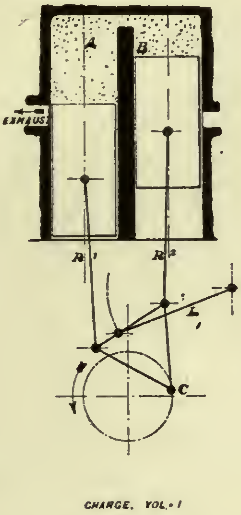


FIG. 115.

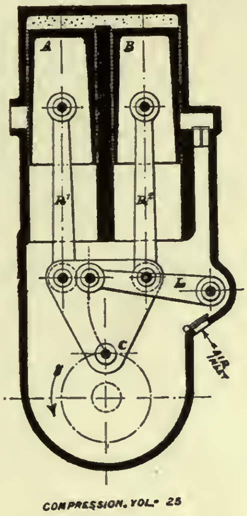


FIG. 116.

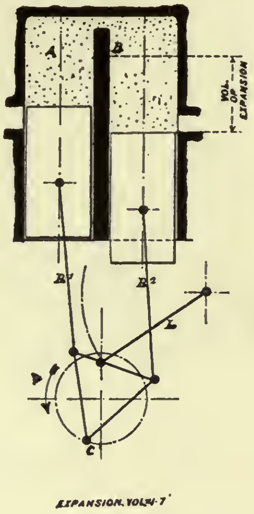


FIG. 117.

Link Motion for Two Cylinders.

The connecting rod, slides, and crank must therefore be carefully designed, with an eye upon the pressures and strains set up, due to the angularity of the link, in this case the connecting rod.

Of course, by good design and excellent material and workmanship it is possible to make a combination of links and pinned joints work satisfactorily. A marine engine has been made to operate by link motions in order to enable two cylinders to run upon one crank pin. This link transmission gear has recently been adapted to gas and oil engines by Mr. F. Lister, Keighley. In this case, however, it has a further advantage. Besides connecting two cylinders to a crank pin, it gives a greater expansion of the gases in a 2-cycle engine. The three Figures (Fig. 115, 116, 117) will explain this link motion and its effect on the gas engine.

The steam engine gear is the same, so that these illustrations will illustrate the gear as applied to any engine.

It will be seen that two cylinders, A and B, are arranged parallel to each other above the crank shaft, A being the exhaust and B the inlet cylinder, connected by a common compression chamber at their inner ends. The pistons are joined by the connecting rods R^1 and R^2 to two corners of the triangular frame, as shown, the other corner being attached to the crank pin C. The movement of the frame is constrained by the radius rod L, the other end of which is jointed to the casing of the engine. Ignition of the compressed charge takes place when the pistons are in the position shown by Fig. 116. The crank rotates in the direction of the arrow, so that piston B travels faster than piston A, and has approached the end of its out-stroke by the time the latter piston has arrived at the exhaust port. Their positions are then as in Fig. 117. When the exhaust port is uncovered the pressure drops to atmospheric, and piston B, then passing an inlet port communicating with the enclosed crank chamber, allows a volume of air to pass through the check valve into the cylinder B in order to scavenge the cylinders from the products of the previous explosion. The piston B then commences its stroke again in advance of piston A, forcing out a quantity of air, and nearing the end of its in-stroke at the time the exhaust port is closed by piston A. The position of the pistons before compression is shown in Fig. 115. Shortly before the closing of the exhaust port a charge of gas or oil is pumped into the cylinder B, forming an explosive mixture with the air previously drawn in. In engines the close governing of which is not essential, the charge may be drawn into the crank chamber with the air, and thence delivered to the cylinder, thus doing away with the necessity for pump charging, though the advantages of scavenging are lost by this arrangement. The mixture is compressed as the pistons approach the upper end of the cylinders. Ignition is effected by any of the usual methods, and the cycle is repeated as before, one explosion taking place to every revolution of the crank shaft.

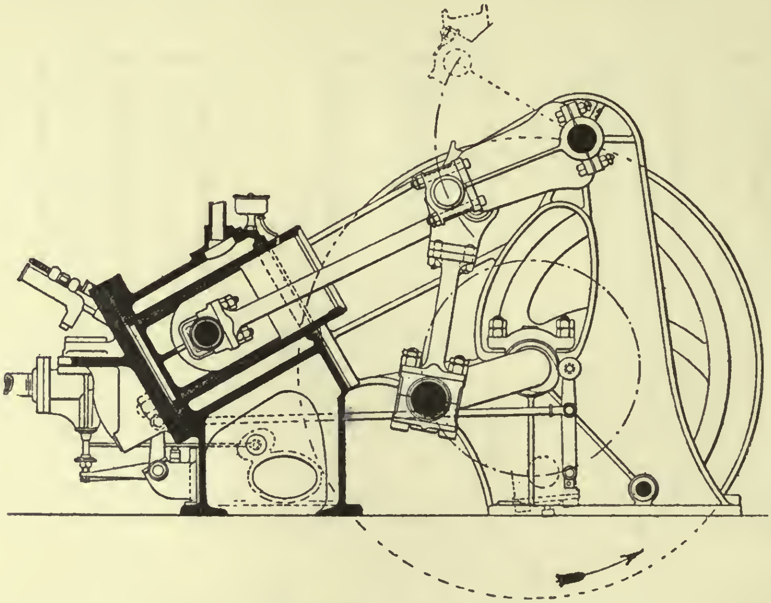


FIG. 118.—Atkinson's Link Motion Engine.

It will be noticed that the initial volume of the charge is increased by from 50 to 70 per cent. before exhaust, allowing more work to be obtained from the fuel, together with a lower exhaust pressure. The ratio of expansion volume to compression volume is 6.8. The design permits of the connecting rods being kept very short, and they are so proportioned that at no point of the stroke do they make a greater angle with the centre line of the cylinders than 5° ; thus the pressure on the cylinder walls and consequent wear is very small. The combined effective power strokes of the two pistons are approximately equal to 1.8 times the crank stroke, the compression portion of the return stroke amounting to 1.2 times the crank stroke.

Another ingenious link motion was invented by Atkinson for the same purpose, but with one cylinder only (Fig. 118).

The four Figs. 119 to 122 show the four positions of the connecting rod, crank, and links. The fixed centre (Fig. 119) is rather below the horizontal centre line of the cylinder, and causes the rod to rise through a greater angle above than below the centre, so that the explosion stroke was 11.1 inches. In the next Figure (Fig. 120) the position at the end of the working stroke is shown. In Fig. 121 the exhaust ends, the link, is inclined instead of horizontal, and thus the in-stroke is lengthened, bringing the piston close to the cylinder end and expelling all the exhaust gases. Next, the piston in Fig. 122 has made a forward stroke, but owing to the small angle made by link below the horizontal this stroke is only 6.3 inches long. A compression stroke inwards is now made, and the cycle started over again.

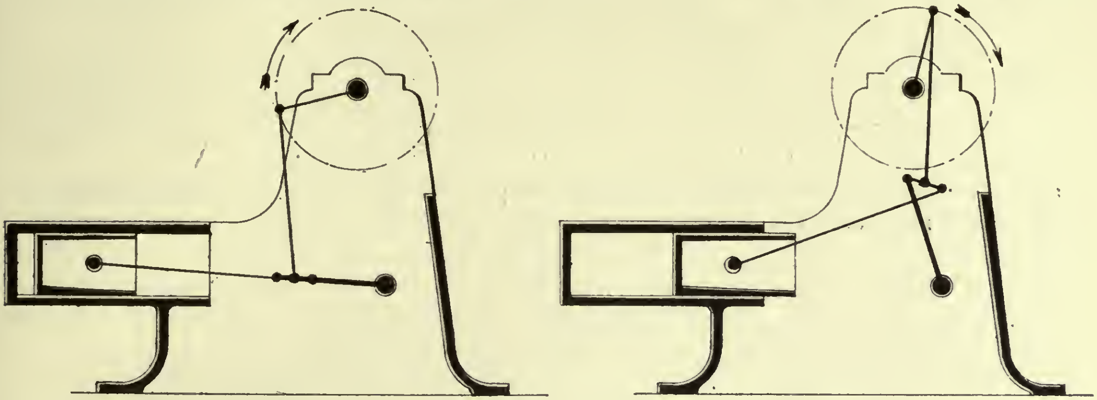


Fig. 119.—Ignition.

Fig. 120.—Expansion.

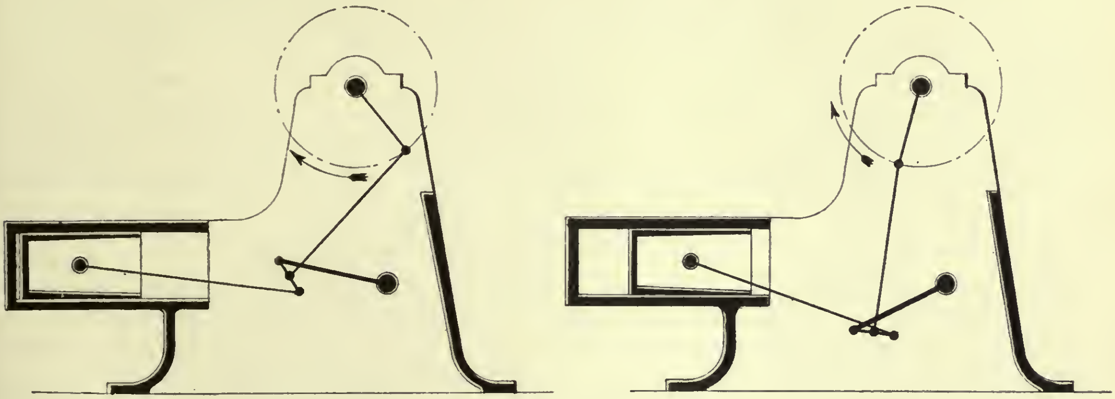


Fig. 121.—Exhaustion.

Fig. 122.—Suction.

Connecting Rod, Crank, and Links.

By means of this link motion the piston makes two double strokes, all of unequal length for each revolution of the crank shaft.

Were it not for the objections to transmitting pressure by links and joints, this is by far the best motion for internal combustion engine, and the advantages are so great that this motion is worthy of study by mechanicians.

The simple connecting rod used in the great majority of engines is shown in diagram, Fig. 123, which explains how to find the relative positions of the crank and piston, and from that also can be obtained the relative position of the slide valve and eccentric. AB is the crank, H the connecting rod cutting the centre line at G in the eye of the cross-head. With G as a centre describe the arc BE, the length AE will be equal to the distance of the piston from the middle of the stroke when the crank is in

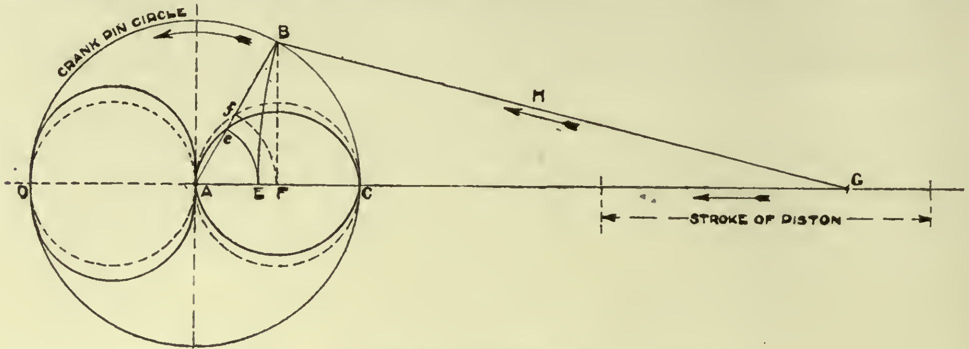

CRANK PIN CIRCLE
B
H
S
e
O
A
E
F
C
G
STROKE OF PISTON

FIG. 123.—Crank and Connecting Rod. Diagram.

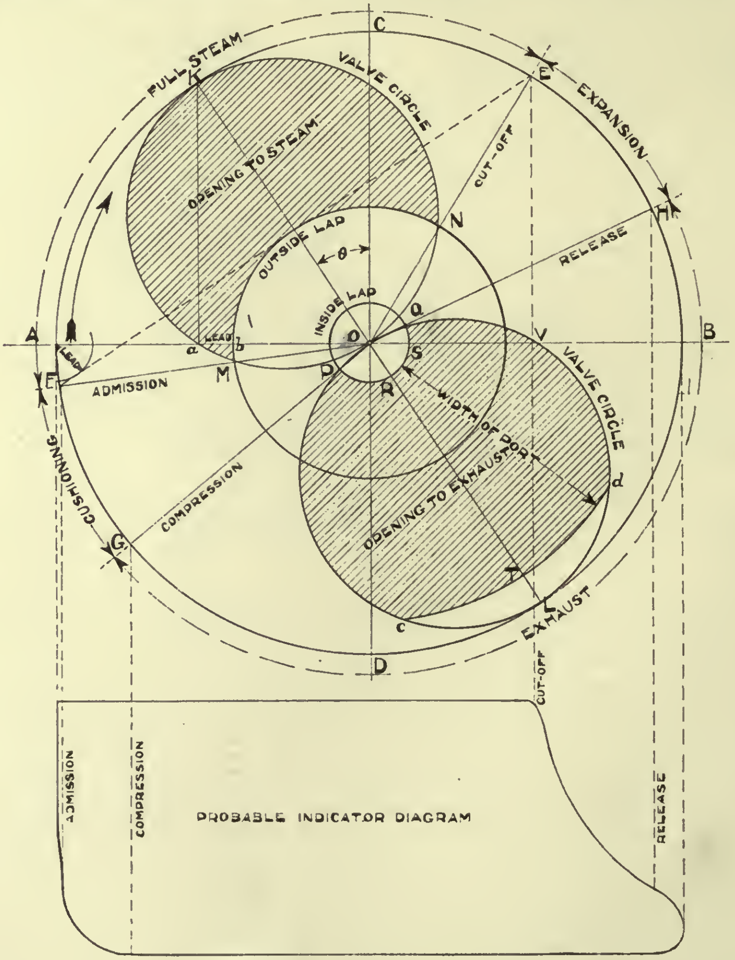

C
FULL STEAM
K
VALVE CIRCLE
E
EXPANSION
OPENING TO STEAM
CUT-OFF
H
N
OUTSIDE LAP
θ
RELEASE
INSIDE LAP
Q
A
I
a
LEAD
b
O
V
B
LEAD
F
M
P
S
R
ADMISSION
VALVE CIRCLE
WIDTH OF PORT
CUSHIONING
OPENING TO EXHAUST
d
COMPRESSION
G
T
L
EXHAUST
c
D
CUT-OFF
ADMISSION
COMPRESSION
PROBABLE INDICATOR DIAGRAM
RELEASE

FIG. 124.—Valve Diagram. Zeuner's.

the position AB. If this distance AE be set off along the crank by drawing with radius AE the arc E*e*, and doing the same for a series of different positions of the crank, all the points will be found to be in the polar curve A*e*C. Any chord of this curve drawn from the point A will be equal to the distance of the piston from the middle of its stroke when the crank lies along that chord.

Valves are seldom set in smaller engines by diagrams. They are in nearly all cases set by the engine fitter setting the crank in various positions, and adjusting the eccentric on the shaft, afterwards keying it to the shaft.

ZEUNER'S VALVE DIAGRAMS

The most common problem in setting out a valve diagram is to find the angular advance of the eccentric and the amount of outside lap required on the valve, with the position of the crank at cut-off given, and a given amount of lead and valve travel.

In Fig. 124 this problem is solved. With centre O describe circle with radius equal to the throw of the eccentric or half the travel of the valve, and through O draw AB, CD at right angles to each other. From O draw OE to cut the circle in the position of the crank at cut-off. With A as centre and radius equal to the lead of the valve, describe a circle, and from E draw a line EF touching this circle. At right angles to EF draw a line KL.

On KO and OL describe the valve circles. With O as centre describe circle with radius from O to point where line EF cuts line KO. This circle is the outside lap circle. O*b* being equal to the outside lap, and *ab* equal to the lead.

If the position of the crank at the point of compression or release is given, draw OG or OH, and where this line cuts the valve circle at PQ describe a circle with centre O, radius OP or OQ. The radius of this circle gives the inside lap.

The diagram, now complete, shows positions of engine crank for various points on the indicator curve. It should be further noted that points F, E, G, and H may be found by the tangents FE and GH, and that the perpendicular distance from A to FE is equal to lead. It is not practically possible to have full opening to exhaust, as given by diagram; so the width of steam port must always be marked off by arc *cd*.

ACTION OF THE CRANK

The turning effort made by a connecting rod and crank varies with the position of the crank. When the centre line of the crank is in line with the centre line of the shaft and cylinder the crank is on the dead centre, and the pressure behind the piston only presses against the bearings without turning the crank. When the crank is at right angles to the line joining the centre of the shaft and cylinder the turning force is a maximum.

With ordinary cranks and connecting rods the turning moment is equal to the total pressure on the piston, multiplied by the distance from the centre of the crank shaft to the point where the centre line of the connecting rod produced cuts the line drawn through the centre of the crank shaft at right angles to the piston's motion.

Fig. 125 is a diagram, the turning moments graphically showing the pressures and positions.

Let $PC = P$, the pressure on the piston to any convenient scale.

The $PD = R$, the direction of pressure along the connecting rod on the same scale.

And $CD = L$, the direction and pressure on the guide slides and blocks.

$$\begin{array}{l} PD \cos\phi = PC. \\ R \cos\phi = P. \end{array} \qquad \therefore R = \frac{P}{\cos\phi}. \quad \text{And } \frac{CD}{CP} = \frac{L}{P} = \tan\phi. \quad \therefore L = P \tan\phi.$$

$$\text{The turning moment} = R \times CF = \frac{P}{\cos\phi} \times CD \cos\phi = P \times CD.$$

From this we can construct the polar curves. Take the crank at any angle to the centre line, say CB, produce the centre line of the connecting rod to cut the line CD in C, with centre C and radius CD describe the arc DK, cutting CB in K. Then K is a point on the tangential pressure curve, and the turning moment for position CB is P × CK. A similar construction can be carried out for all other points, and a complete curve made by joining the different points. This curve is not a circle, neither is it a regular wave motion. The curve rises abruptly during the first quarter of a revolution, falls flat during the second quarter, thus indicating a greater pressure on the crank during the first quarter, and it shows also the maximum pressure is reached before half-stroke. In practice, however, we must take into account that the pressure on the piston decreases from the point of cut-off, and also that inertia must have considerable effect on the crank pin pressures. All the reciprocating parts of the engine connected with the piston must be brought to rest and started in motion in the reverse direction twice every revolution.

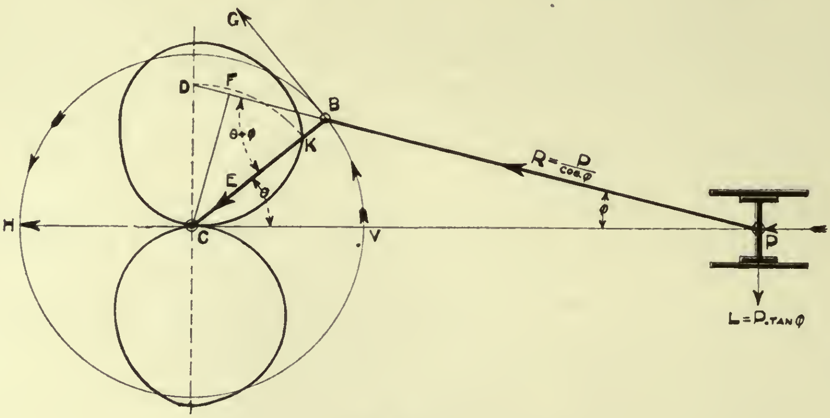


FIG. 125.—Crank and Connecting Rod.

To calculate the pressures on the crank pin at the beginning and end of the stroke, the mechanical formulæ $\frac{mV^2}{r}$ gives the centrifugal force where V is velocity in feet per second, r radius of circle in feet, and m units of mass.

But we measure mass by weight, and take revolutions per minute in a circle of feet radius, and hence the formulæ becomes $m = \frac{W}{g} = \frac{W}{32}$, where W = lbs., n = revolutions per minute. Then $\frac{mV^2}{r} = \frac{W}{gr} \cdot \left(\frac{Q\pi rn}{60}\right)^2 = 0.000341\ Wrn^2$, the well known formulæ for centrifugal force.

For example, the stroke of the engine is 4 feet (crank 2 feet radius = r), W = 1256 lbs. The weight of the moving reciprocating mass, and n the revolutions per minute = 50. Hence $0.000341 \times 1256 \times 2 \times 50^2 = 2141$ lbs.

To balance this pressure cushioning is adopted, that is, the exhaust is cut off and compressed at the end of the stroke to such a pressure as will slightly more than overbalance this force. The pressure required is found by dividing the acceleration, in this case 2141 lbs., by the piston area. Suppose the piston area 214 square inches, then $\frac{2141}{214}$ = 10 lbs. pressure as a steam cushion required to relieve the crank of the effect of inertia.

It is usual nowadays to take piston speed in feet, and that value is often given without reference to n, or length of stroke L, and may be represented by a symbol E, and E is equal to $\frac{Q\pi nr}{60}$.

Accelerating force is then equal to $\frac{mE^2}{Qg}$, where m is weight of reciprocating mass E, the speed in feet per second of the crank pin or piston, and $g = 32$.

Engines are generally run up to 800 feet per minute; piston speed, 13.3 feet per second. Some take as a standard 12 feet per second, which gives an easily remembered formulæ for inertia per lb. of reciprocating mass $\frac{144}{64} = \frac{9}{4} = 2.25$ lbs. per lb. of reciprocating mass at a standard speed of 720 feet per minute.

FLY-WHEELS

It will be seen from the foregoing that the crank effect is far from regular. At one part of the stroke it is very great, at another very small or negative when the cushioning takes place.

By calculating out the turning moments at the maximum and minimum crank efforts we can find the surplus and deficit of energy at the two points. The object of the fly-wheel is to absorb the surplus by going very slightly faster at that point of surplus, and slower over the region of deficit; hence a fly-wheel must vary in speed to some extent. And this variation has been found of practical importance in running alternating dynamo electric generators in parallel.

The heavier the fly-wheel the smaller the variation.

The diameter of the wheel is found by a formulæ where D is the diameter, and N the revolutions per minute. The greatest diameter for solid cast-iron wheels is given by $D = \frac{1528}{N}$, and for wheels built in segments $D = \frac{955}{N}$.

If N_2 is equal to the maximum revolutions, and N_1 the minimum, and N the mean number, then the co-efficient of fluctuation $K = \frac{N_2 - N_1}{N}$.

K in practice varies from 0.01 to 0.002; the latter for electric alternators.

The weight of the fly-wheel is calculated by first finding E, the excess of energy, at the maximum of the stroke over the minimum energy at the lowest pressure of the stroke.

W = Equals weight of fly-wheel rim in pounds.
V = Mean velocity feet per second.
K = Co-efficient of fluctuation.
g = 32.

Then $W = \frac{g \times W}{V^2 \times K}$.

100 feet per second is the highest limit of speed for the periphery of a fly-wheel.

STEAM PORTS

The velocity of steam through steam ports should not exceed 140 feet per second, but 100 feet is the usual limit. The width of the steam port is about .75 of the cylinder bore in most cases. The length = L of opening is then found by formulæ

$$L = \frac{AS}{60 \times WV}.$$

Where—A = Area of piston in inches.
S = Piston speed in feet per minute.
W = Width of port in inches.
V = Velocity of exhaust flow in feet per second.

BALANCING OF ENGINES

This important subject is now better understood, and engines are now much better balanced than formerly. The tons of masonry and holding down bolts are not relied upon to cover the defects of unbalanced engines. In reciprocating engines the rotary parts can be balanced. And so also to a large extent can the reciprocating parts.

In steamships it is of the utmost importance to obtain good balance to prevent vibration of the ship. Hence one great advantage of the steam turbine. One system of balancing consists in using bob weights, as has been done by Mr. Yarrow, and shown in diagram Fig. 126 on one cylinder only. The weights are carried on rods upon cranks or eccentrics, so that they travel up and down in opposite directions to the piston, and so counteract the weights.

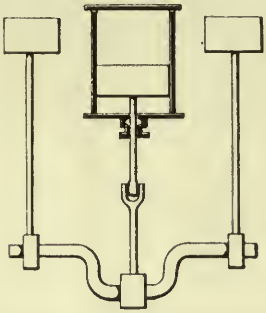


Fig. 126.—Balancing Engine.

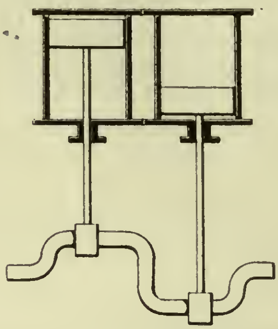


Fig. 127.—Balancing Engine. Single Acting.

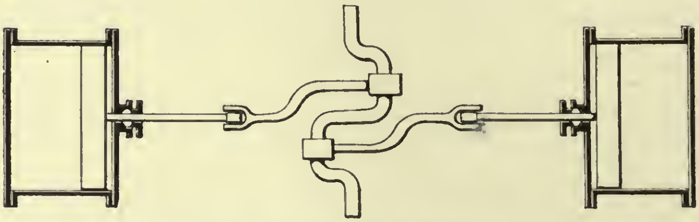


Fig. 128.—Balancing Engine.

Two cylinders side by side on opposite cranks nearly balance, but still there is a couple between their cylinder centres which cause vibration. This is shown in diagram Fig. 127. Two cylinders opposite each other make a fairly good balance, as shown in Fig. 128. Here the two pistons move oppositely and nearly in the same line, so that no couple exists. There is, however, a tendency to vibrate, due to the weights moving from the side to the centre if the engines are in a ship and athwart the ship.

The best balance is that obtained by a 3-cylinder engine with the cylinders all in same plane at an angle of 120° to each other, and on one crank as shown in Fig. 129 in diagram. In all probability the reciprocating engine will give way entirely to the turbine on steamships, so that balancing will not be necessary in the future.

Admiral Melville, of the U.S. American navy, has contributed a considerable amount of information on balancing marine engines. From his paper the following details of a Macalpine engine are given.

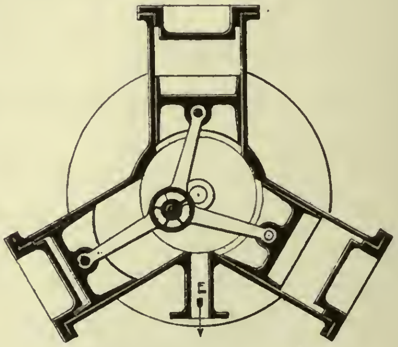


Fig. 129.—Balancing Engine.

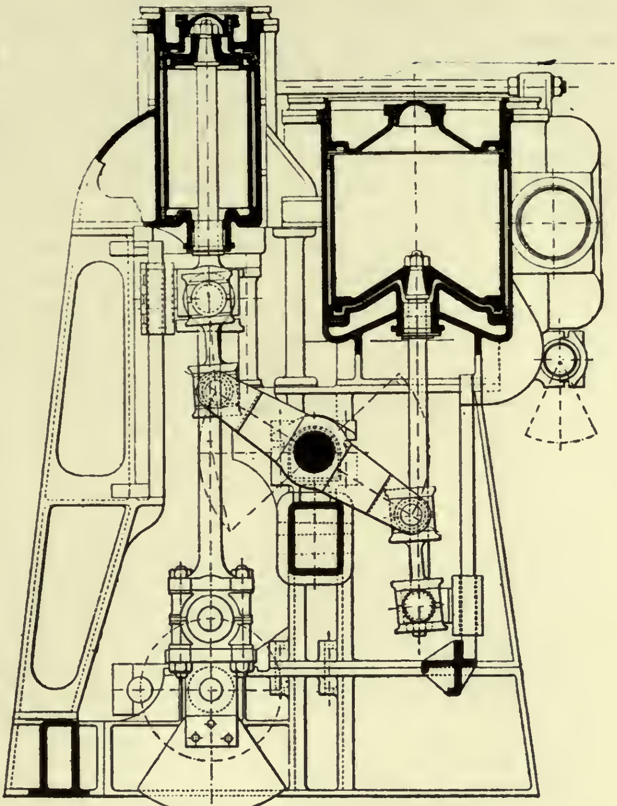
Fig. 130 shows a 4-cylinder quadruple engine, working on two cranks. The high-pressure and first intermediate-pressure cylinders are immediately over the cranks.

The second intermediate-pressure is connected through levers and links to the high-pressure cross-head. Similarly, the low-pressure is connected to the first intermediate-pressure cross-head. It is at once evident that the upward movement of the high-pressure piston is precisely the same as the downward movement of the second intermediate-pressure. By proper adjustment of the moving masses a perfect vertical and horizontal balance can be attained; for a comparatively light counterbalance on the crank exactly balances the revolving masses vertically and horizontally. The levers and links can be so adjusted that there is no unbalanced horizontal force from them, and it only remains to properly adjust the masses of the parts moving with the cross-heads to make the total vertical force zero.

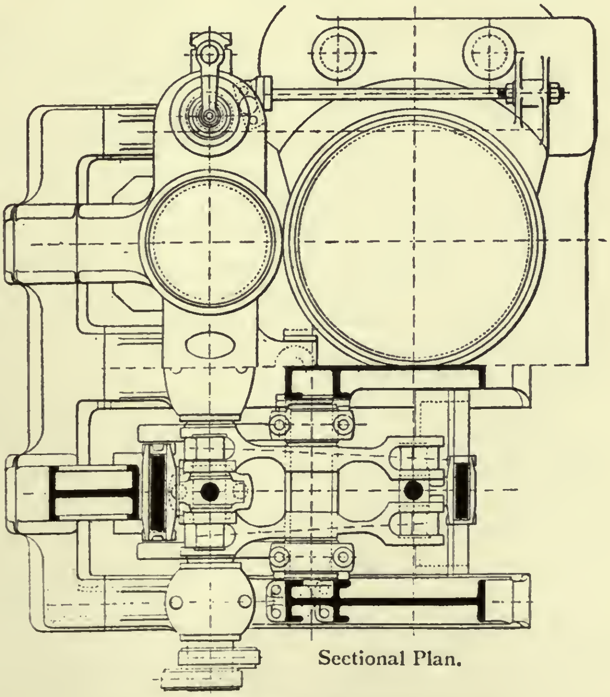
To make clear the mechanical advantages of this design the Admiral states:—

"The bearing surfaces in this and the merchant engine design have all been increased. Though the surfaces in the earlier design seem ample, the increase puts the long running of the engine without readjustment beyond question.

"By bringing the main guides to the outsides their accessibility is considerably increased, and it also allows the levers, though raised, to be kept closer together with their centre bearings outside them and in easy reach. Making a centre front column leaves the engine perfectly open over the main bearings, so that not only these, but also the main cross-heads, front links, and centre lever bearings, can be easily got at, even when the engine is running. To make this easier at the after end, the after back column of the



Sectional Elevation.



Sectional Plan.

Fig. 130.—Macalpine's Balancing Engine.

second intermediate-pressure cylinder is moved a few inches aft of the centre of the corresponding main bearing. As strength is easily provided, this shift may be made without fear. Also, to give better head room above the intermediate platform, the high-pressure and first intermediate-pressure valves are worked from above. This is also improved by dispensing with the jaw guides of these gears, as shown in the view of the valve gear (Fig. 131).

"The extra length of back piston rod required by the modification of the design is 25 inches, but this weight is almost entirely saved by the shortening of the links. In any case its inertia at full speed only increases the pressure on the main bearings by 2½ lbs. per square inch, so that the addition is unimportant.

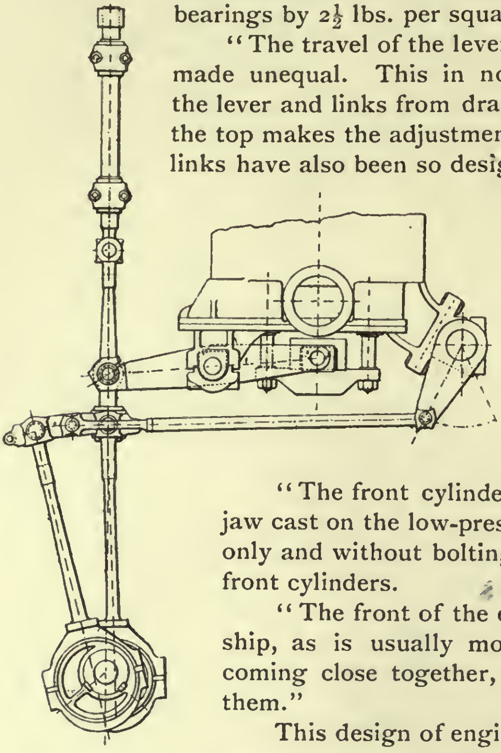


FIG. 131. — Valve Gear Balancing Engine.

"The travel of the levers above and below the horizontal position is made unequal. This in no way affects the balance, but by preventing the lever and links from drawing so nearly into line when the crank is at the top makes the adjustment of the whole connection less delicate. The links have also been so designed that they can be kept exactly to length. The levers can readily be withdrawn to the back. The back links and glands of back cylinders are in easy reach.

"The cross ties are attached so that the expansion of the cylinders will have no sensible effect.

"Some of the weight saved by bringing down the back cylinders is added to the front columns and some to the front of the bed-plate, so that the total is not much altered.

"The front cylinders are held from fore-and-aft movement by a jaw cast on the low-pressure cylinder, which holds, by sliding contact only and without bolting, the foot for the centre back column of the front cylinders.

"The front of the engine may now be toward the centre of the ship, as is usually most convenient; and the hatches, instead of coming close together, will leave a wide passage on deck between them."

This design of engine incidentally has an advantage in taking up much less room in the ship. The non-vibrating engine is also necessarily the better and safer to run.

The engine, as shown in Fig. 130, has an unbalanced inertia torsional couple. By placing the cylinders tandem, or adding a third cylinder on the other side of the vertical line through the shaft, and also connecting it with a lever, any increase of unbalanced torsional couple over that in ordinary forms of engine is avoided. But these arrangements are not to be commended for ship-board.

Macalpine long ago pointed out that by increasing the number of cranks almost perfect torsional balance can be attained. Take three cranks at 120°, and make the moving masses alike for each of the three cranks. The first-period torsional couple for the forward crank may then be represented by $C_1 \cos \theta$; and the total torsional couple from the three cranks will be

$$C_1 \cos \theta + C_1 \cos (\theta + 120^\circ) + C_1 \cos (\theta - 120^\circ) = 0.$$

Similarly, for second and fourth-period torsional balance, we have—

$$C_2 \cos 2\,\theta + C_2 \cos 2\,(\theta + 120^\circ) + C_1 \cos 2\,(\theta - 120^\circ) = 0.$$
$$C_4 \cos 4\,\theta + C_4 \cos 4\,(\theta + 120^\circ) + C_4 \cos 4\,(\theta - 120^\circ) = 0.$$

Similarly, the valve gears for each crank may be balanced. This gives perfect torsional balance for first, second, and fourth periods.

For very large engines it would probably be advisable to adopt three cranks and six cylinders. In such cases it is usual to decrease the size of the largest cylinders by increasing their number.

With four cranks in opposite pairs, at right angles, we could only have first and second-period torsional balance.

INSTRUMENTS FOR TESTING STEAM ENGINES

Watt scientifically, from the very first experiments, examined the problems of the steam engine, and this fact is most strikingly shown by his invention of the steam engine indicator, whereby he could see how the steam pressures varied in the cylinder. It showed also how the valves acted, how the steam behaved at admission during the stroke and at exhaust ; and most important, it gave a means of ascertaining the mean pressure on the piston, and from that he could calculate the "indicated horse-power."

The term "horse-power," usually written H.P., is equal to 33,000 lbs. lifted 1 foot high per minute, or 550 lbs. 1 foot high per second.

It is the amount of work which a fair specimen of the horse in Watt's day could be reasonably expected to accomplish. Horses then had evidently not developed in muscular power to the extent of the powerful draught-horses of to-day, some specimens of which can be seen daily in railway lorries walking off with a load exerting three or four times the horse-power adopted by Watt. The early motor car people found this out to their cost. Instead of a 1 horse-power engine running off with a 1 horse-power gig or dog-cart, they found it took a 4 or 5 horse-power engine to do the work of the one horse ; even a motor bicycle takes about a $1\frac{1}{2}$ horse-power engine. A railway lorry, such as one big horse walks about with, loaded with about 3 tons or so, would be fitted with a 10 horse-power motor for the same work. The electric tramcars on street railways could possibly be drawn by four good conditioned horses ; they are fitted with two 15 horse-power motors. The amount of work in 33,000 lbs. lifted 1 foot high is therefore only, after all, an arbitrary unit, and is not really a "horse-power" at all, and any other unit might just as well have been adopted ; in fact, it would have been well had the unit of power been changed twenty-five years ago when the electrical units were being established. 33,000 foot-lbs. is an awkward figure, but there it is, and it is now so much woven into text-books, tables, and calculations that it would be very difficult to change it at this date. The mechanical equivalent of heat unit is of the value of 778 foot-lbs., equivalent to the heat required to raise 1 lb. of water 1° Fahr.,—called a "British thermal unit," and represented by symbol $\ominus$. This amount of heat, supplied at the rate of one $\ominus$ per second, is equal to 1.414 horse-power if converted into work, and if at the rate of one $\ominus$ per minute is equal to 0.02357 horse-power.

Then, when we come to convert the mechanical into electrical units we find one "ampère" of current at one "volt" pressure has one "watt" of power. A watt is a unit equal to 44.2 foot-lbs., and 746 watts per minute = 1 horse-power.

The Board of Trade electrical power unit is 1000 watt-hours = ampères × volts × time of flow ; say, 5 ampères for 10 hours at 20 volts,—$5 \times 10 \times 20 = 1000$ watt-hours. The horse-power equivalent to this 1000 watt-hours is $\frac{1000}{746} = 1\frac{1}{3}$ H.P.

All these odd numbers and fractions, which so much impede the calculator, are due to the injudicious selection of the fundamental practical units, such as 33,000 foot-lbs.

The horse-power of an engine depends on the dimensions of the cylinder and the piston speed, and in Watt's time an empirical unit was used called a "nominal horse-power."

With the steam pressure then in vogue, namely, 7 lbs. per square inch, at the usual speed per minute, a certain area of piston was found necessary for each horse-power, and engines were accordingly rated by the number of times their piston contained this area.

As the pressure was increased it followed that the engine would give off proportionately greater power, and that the nominal power did not represent the actual power. The original allowance was therefore modified for land practice, and 10 to 11 circular inches of piston area per horse-power was fixed as a convenient figure.

With the continued increase in boiler pressure and much higher piston speed now general this figure bears no definite proportion whatever to the power the engine will exert, and is consequently modified by each maker. The term "nominal horse-power" is accordingly of so little service that it has been practically abandoned by engineers, except in certain classes of smaller engines, where standard sizes are more or less fixed.

Indicated horse-power (or I.H.P.) is generally the most convenient standard, and is also a test which every buyer can from time to time apply. It is the power developed by the steam in the cylinder, and is measured by an instrument attached to the cylinder called an indicator, which marks on a card during the stroke the varying pressure of the steam (Fig. 132).

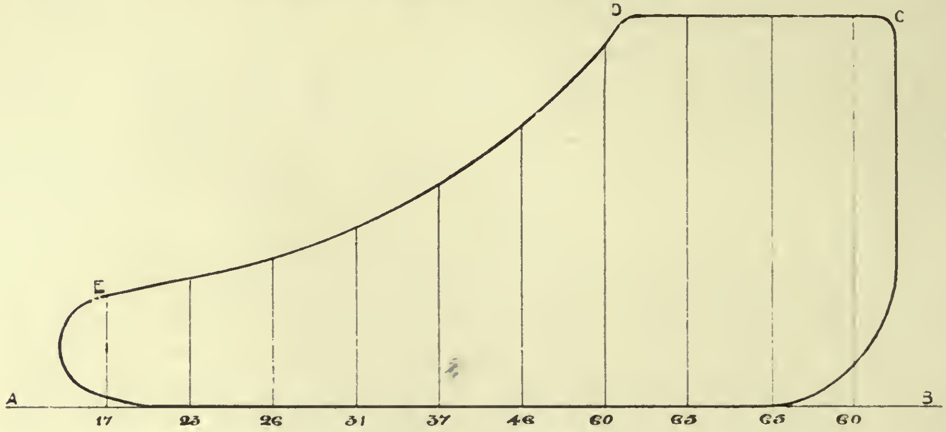


FIG. 132.—Indicator Card.

From this card is obtained the average or mean pressure per square inch in the cylinder, written shortly M.E.P. ("mean effective pressure"), which, multiplied by the area of the piston and the piston speed in feet per minute and divided by 33,000, gives the indicated horse-power.

As the indicated horse-power, however, makes no allowance for the power absorbed in moving the engine parts, it does not represent the actual power given off by the fly-wheel or crank-shaft, which is termed "effective horse-power" (E.H.P.), or often the "brake horse-power" (B.H.P.). In engines such as portables this is most conveniently measured by a friction brake applied to the fly-wheel. In amount it is from 8 to 15 per cent. less than the indicated horse-power; this difference representing the power absorbed in the engine itself.

In engine testing the first thing, then, to know is the indicator diagram; first, in order to see how the valves are set, where the cut-off takes place, where exhaust commences, the compression begins, and so on. If on first taking a diagram all these points are not shown to be satisfactory, the valves must be altered or reset until the diagram is of the proper form. After this has been done it will be correct to take the area of the indicator diagram in conjunction with the strength of spring employed as a measure of the mean effective pressure.

The mean effective pressure multiplied by the piston speed in feet and the area of the piston and divided by 33,000 gives the I.H.P. ("indicated horse-power)." This may be written shortly where P = the mean effective pressure in lbs. square inch ; A = the area of piston in inches, and N = the piston speed in feet per minute,— $\frac{PAN}{33000}$ = I.H.P. A and N are known quantities, or easily measured—A by a foot rule, N by a speed indicator or counter. P must, however, be measured by the indicator, and as it is a most important instrument, some forms shall now be described.

With old-fashioned engines with their slow speeds a somewhat primitive indicator gave good results.

Its earliest form is described by Rankine in his steam engine. It is interesting as it is practically that used by Watt, and it is represented by Fig. 133. AB is a cylindrical case. Its lower end A contains a small cylinder fitted with a piston, which cylinder, by means of the screwed nozzle at its lower end, can be fixed in any convenient position on a tube communicating with that end of the engine cylinder where the work of the steam is determined. The communication between the engine cylinder and the indicator cylinder can be opened and shut at will by means of the cock K. When it is open the intensity of the pressure of the steam on the engine piston and on the indicator piston is the same, or nearly the same.

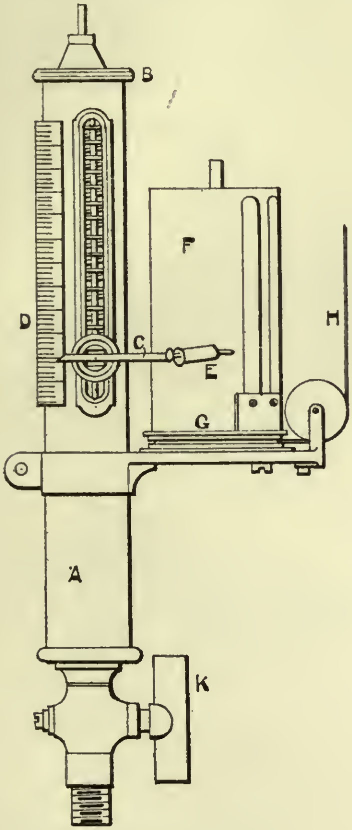


FIG. 133.—Watt's Indicator.

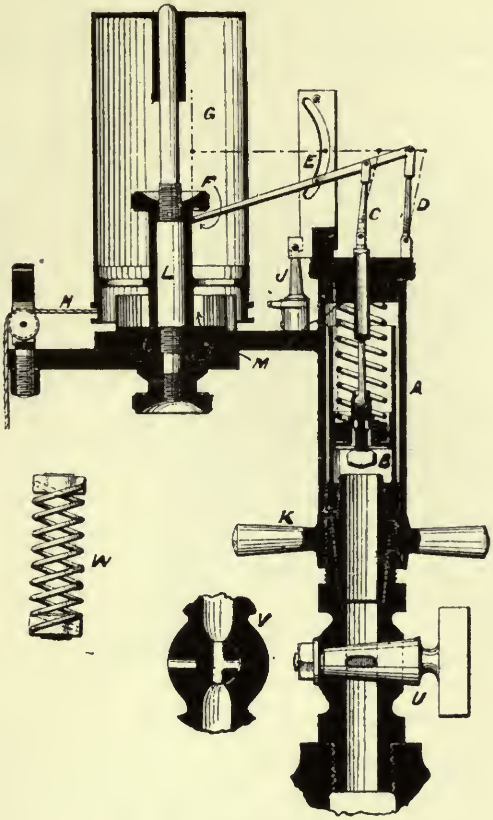


FIG. 134.—Tabor Indicator.

The upper end B of the cylindrical case contains a spiral spring, one end of which is attached to the piston or to its rod, and the other to the top of the casing. The indicator piston is pressed from below by the steam, and from above by the atmosphere. When the pressure of the steam is equal to that of the atmosphere the spring retains its unstrained length, and the piston its original position. When the pressure of the steam exceeds that of the atmosphere the piston is driven outwards, and the spring compressed; when the pressure of the steam is less than that of the atmosphere the piston is driven inwards, and the spring extended. The compression or extension of the spring indicates the difference, upward or downward, between the pressure of the steam and that of the atmosphere.

A short arm C projecting from the indicator piston rod carries at one side a pointer D, which shows the pressure on a scale whose zero denotes the pressure of the atmosphere and which is graduated into pounds on the square inch both upwards and downwards from that zero. At the other side the short arm has a longer arm jointed to it, carrying a pencil E.

F is a brass drum, which rotates backward and forward about a vertical axis, and which, when about to be used, is covered with a piece of paper called a "card." It is alternately pulled round in one direction by the cord H, which wraps on the pulley G, and pulled back to its original position by a spring contained within itself. The cord H is to be connected with the mechanism of the steam engine in any convenient manner which shall ensure that the velocity of rotation of the drum shall at every instant bear a constant ratio to that of the steam engine piston: the back and forward motion of the surface of the drum representing that of the steam engine piston on a reduced scale.

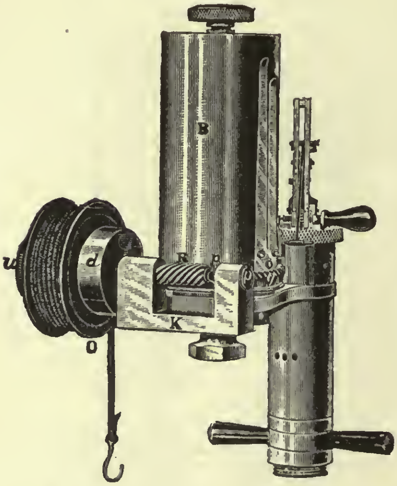
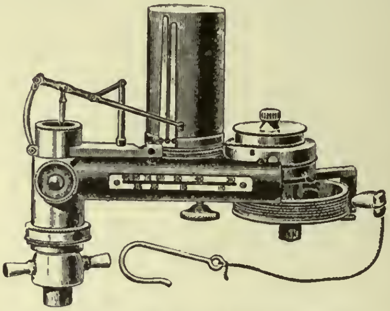


FIG. 135.—Reducing Wheel.



Bachelor Indicator with Reducing Gear.

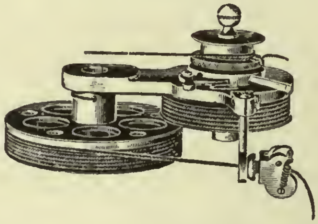


FIG. 136.—Indicator Reducing Wheel.

A good example of indicator is the Tabor indicator (Fig. 134). The great point in all of them is to reduce the inertia of the moving parts, the piston, the piston rod, the links, and pencil arm, and yet have them rigid enough to prevent vibration.

In this indicator the pencil point is compelled to follow a straight line by the curved guide E, which compensates for the radical motion of the arm. It, however, also tends to steady the motion. The indicator is usually attached to the cylinder by a 2-way cock shown at V and U, so that either end of the cylinder may be indicated separately. Various strengths of springs are supplied so as to suit different steam pressures. The indicator is coupled to the cock by a union K for rapid manipulation. The cylinder has an air jacket, which prevents radiation of heat and tends to maintain an even temperature.

In slow-speed engines the horizontal motion of the piston is reduced usually by levers. At high speeds these levers introduce vibration and inertia, and it is difficult to hitch on the cord. The Tabor indicator has been made with a wheel and worm reducing gearing, which to a large extent overcomes this objection. It is shown in Fig. 135. By altering the pulley O various sizes of diagrams can be taken. The cord is directly attached to the piston cross-head, and travels the whole stroke, a spring in barrel D winding it up again on the return stroke.

The Bushnell reducing gearing and cord take-up are shown in Fig. 136. It can be adapted to any indicator. A separate gear for attaching to any indicator is shown in Fig. 136, and it is also shown attached to the Bacheldor indicator. This indicator has a flat spring, the tension of which is adjustable by a clip movable along the horizontal tube, and a scale indicating the valve of the spring at various points ; hence one spring serves for a considerable range of pressure. These cord gears have entirely superseded the old-fashioned waggling levers.

A still further improvement has been made in the Wayne indicator (Fig. 137), in which the levers and joints have been abolished in the indicator itself, and the movement of the working parts very much reduced, and the opposing spring is more accessible.

Instead of a cylindrical piston a rotary piston is used, shown in section in Fig. 138. *e*, *e* are exhaust ports to allow of the escape of any steam and water leakage. The pencil vibrates in a circle, so that the cord is curved to the radius of the pencil arm. The card is carried on an aluminium frame, which is pulled horizontally by the cord through ordinary reducing gearing. The opposing spring to the piston is outside, being a spiral on the spindle end opposite the arm.

For indicating very high-speed engines the Wayne indicator may be used to draw "line" diagrams. By this gear the movement of the piston is reduced to $\frac{1}{20}$ of an inch at a time, and the diagram drawn a little bit at a time line by line. By turning the crank handle a worm is rotated which gears with a worm wheel in a concentric cylinder with the working cylinder. A radial arm fixes the worm wheel to the piston rod by a screw in a slotted hole. The slot allows of $\frac{1}{4}$ inch motion of the pencil when the worm wheel is stationary.

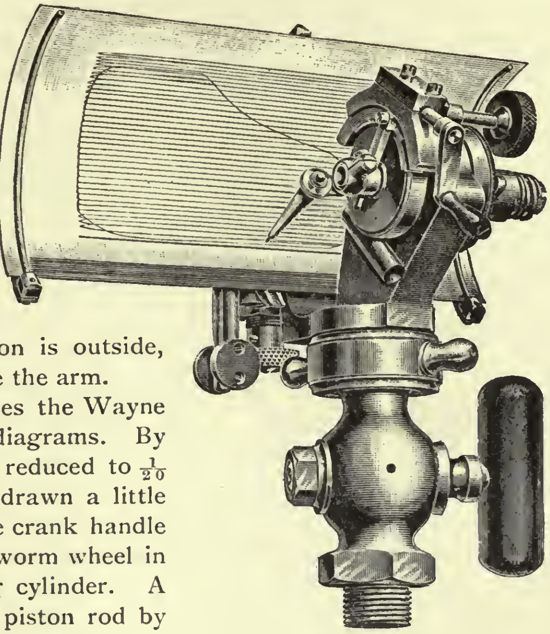


FIG. 137.—Wayne Indicator.

To use the instrument thus, turn the crank handle until the slot in the radial arm stands free of the screw. Then take the atmospheric line, turn on the steam, and commence to turn the crank handle. Now, as the card reciprocates lines will be drawn horizontally, and their distance apart will depend on the loss of time caused by the slot in the radial arm, and consequently on the speed of turning the crank handle. It is an extremely efficient and handy machine, and for high speeds almost indispensable.

To find the mean pressure the diagram (Fig. 132) may be divided up into 10 along the atmospheric line, then measure the height of each line by the scale corresponding to the spring employed. By adding up the heights and dividing by the number of lines we get approximately the average pressure. But the most correct method is to employ a "planimeter" or "averager," a machine which gives the area of the diagram, and from that and the scale the mean pressure is at once correctly found. In the Coffin instrument the diagram to be measured is placed between the two vertical straight edges. The wheel running on a smooth piece of paper, the pointer is then started at any point marked on the diagram, carried round the diagram back to the starting point in a clockwise direction. The wheel and vernier scale will then read off the area. To get the mean pressure from this, we move the pointer straight upwards against the straight edge until the wheel scale comes to zero. Say the distance moved up is EA, then this height is the mean height, which, measured by the spring scale, gives the mean pressure.

This distance EA may be small, so that an error in measurement of a slight amount may mean a large pressure error. This may be to some extent avoided by taking the pointer 3 times round the diagram, the distance EE will then be 3 times the mean. It can be measured and divided by 3 to get the actual mean, and being a long line a slight error is not of so much value.

There are other indicators of the diagram class, and one which gives the mean pressure directly, thus performing both the indicator and averager operations in one. It was first introduced by Professor Boys, of South Kensington Science College, and has since been introduced in improved form by the name of Little's indicator (Fig. 139).

It depends upon the principle that if we press a wheel edge upon a rotating drum the wheel will be driven round at the same peripheral speed as the drum if the two axes are parallel; but as we incline the axes of the wheel to an angle with the axis of the drum the wheel will be driven at a less speed until when the axes are at right angles the wheel will have no speed at all.

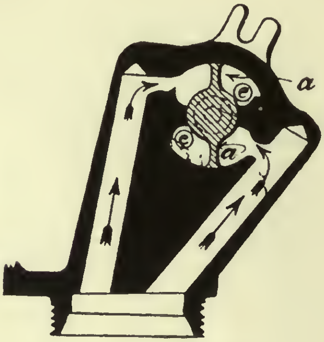


FIG. 138.—Wayne Piston.

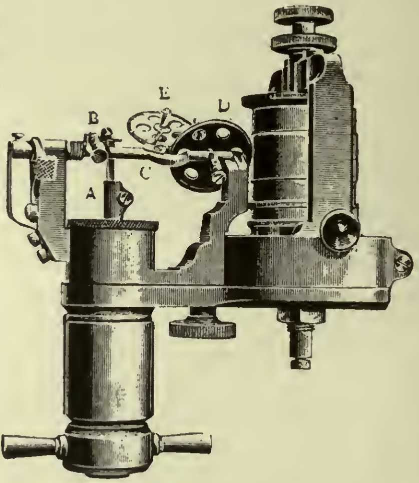


FIG. 139.—Integrating Indicator.

The Little integrating indicator (Fig. 139) consists of a wheel D, which can be inclined at an angle with the axis of the card drum of the indicator against which it presses. It has a scale wheel and vernier, same as in Coffin's averager shown at E, and the wheel is inclined by any of the ordinary forms of spring indicator pistons.

The length of stroke of the drum must not exceed 2 inches, and the scale of reading is one-tenth. The reading of the instrument I multiplied by 10 and divided by N, the number of strokes per minute, gives the mean area $\frac{I \times 10}{N}$; and the mean height is equal to $\frac{I \times 10}{N \times 2}$, 2 being length of drum stroke, and the M.E.P. $= \frac{I \times 10 \times \text{scale of spring}}{N \times 2}$. It can be worked for hours at a stretch, and so give the mean pressure over a period of time.

Professor Boys also proposed a form of indicator which has some interest. It consisted of a cylinder piston and spring, with an indicating pointer like a pressure gauge. By means of a long thin tube the steam by friction in the pipe cannot move rapidly, so that the piston stands still at a height proportional to the mean pressure. I made many experiments with this type of indicator, and found it very accurate. But this accuracy was obtained after a long series of tests on pipes of different bore and lengths. It was

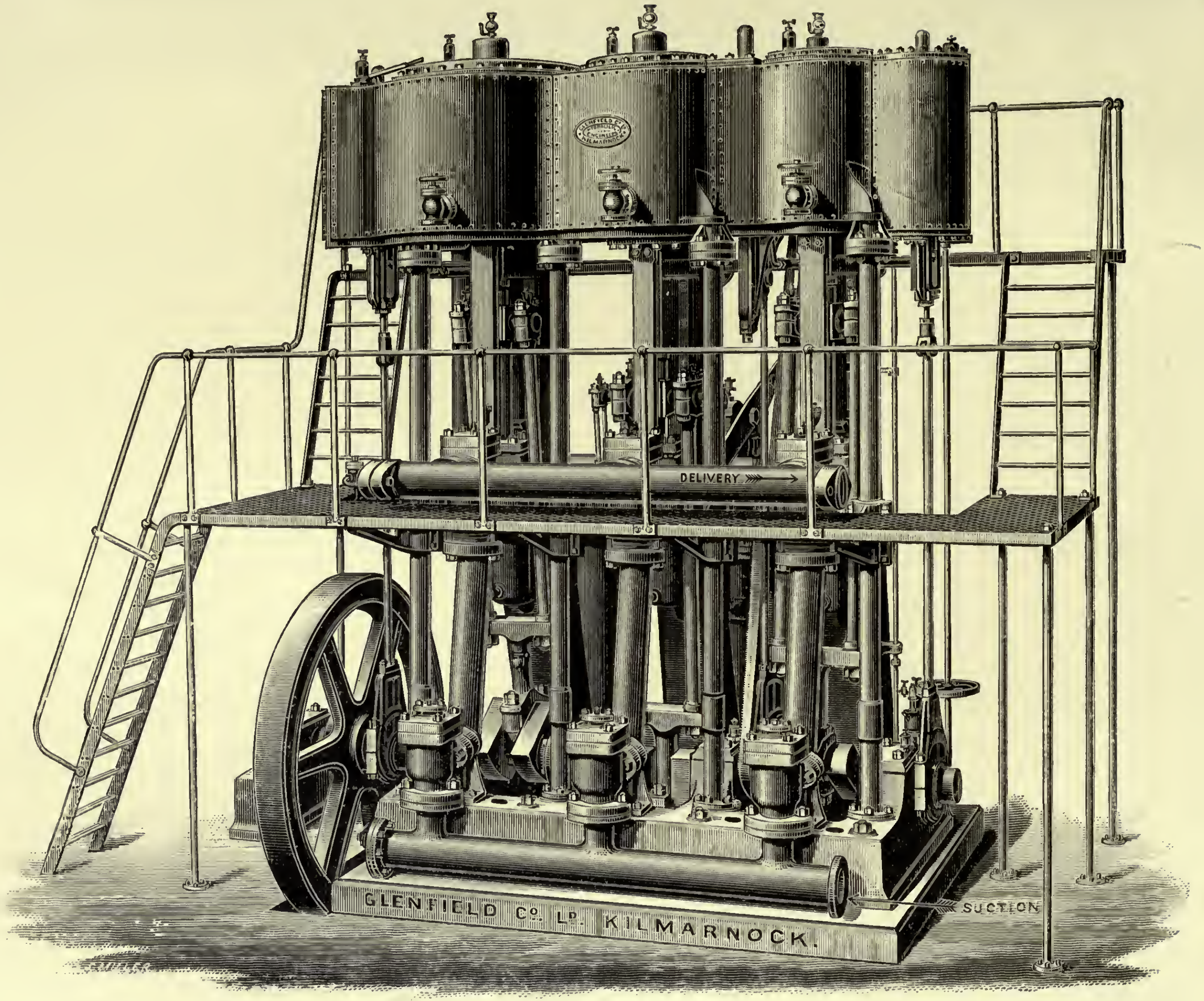

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PLATE III.—TRIPLE-EXPANSION VERTICAL PUMPING ENGINE.

found that for every different bore there is a corresponding length at which the passage of the steam is absolutely proportional to the difference of pressure between the two ends of the pipe.

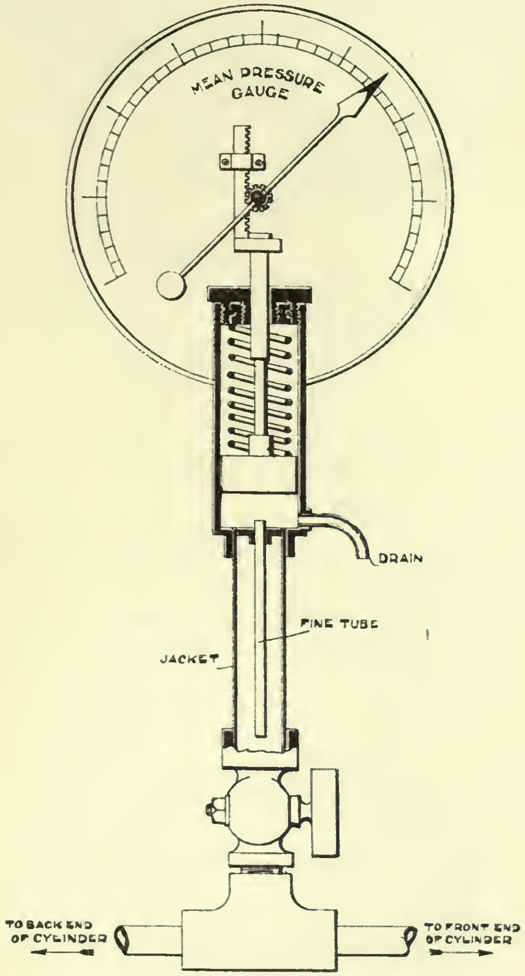


Fig. 140.—Rankin Kennedy's Mean Pressure Indicator.

This was found by means of a balance, on one end of which was hung an inverted vessel (Fig. 141), on which was fixed the pipe under trial. The vessel dipped into another containing water, so that by displacement the air could be driven out of or drawn into the vessel through the pipe by a known pressure produced by a weight placed on the opposite scale pan, or on the inverted vessel. The oscillation of the beam was measured between two stops making electric contacts and ringing a bell, and the time of the oscillation taken by a stop-watch.

By means of this apparatus a tube could be adjusted in length,—by taking it long enough to start with and cutting it down until the air discharge or ingress was proportional to the difference of weight on the two ends of the beam.

When the right length has been thus obtained it will be found that with a certain bore of tube and a difference of weight of 1 ounce the time of an oscillation will be, say, 30 seconds, with ½ ounce it will be 60 seconds, with ¼ ounce 120 seconds, and so on.

An indicator on this principle (Fig. 140) is very good on high-speed double-acting steam engine cylinders. The chief difficulty has been to design it so that water could not condense in or lodge in the fine tube. The best type has the fine tube steam jacketed, and a fine orifice in the bottom of the cylinder whereby a discharge of steam takes place, carrying with it any water condensed. The cylinder is also jacketed like the Tabor indicator. When well constructed it gives a continuous indication of the mean pressure, and in conjunction with a clock and drum recorder gives a record of mean pressure during a day's run.

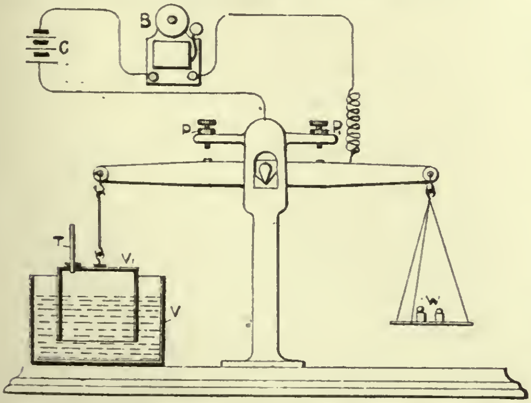


Fig. 141.—Rankin Kennedy's Tube Testing Balance.

It is important to take the speed of the engine correctly, and if there is any suspicion about steadiness of running, a tachometer or speed indicator is the best instrument. It is shown in Fig. 142. It is driven by belt from the engine shaft from a pulley with a known ratio of diameter to that on the speed indicator. This drive should be thoroughly reliable, and the readings carefully checked by a stop-watch and speed counter.

A speed counter of simple construction is shown in Fig. 143. The square centre runs freely without driving the counter until the moment when counting is to begin. It is put into gear by slight pressure, and stops instantly on withdrawal.

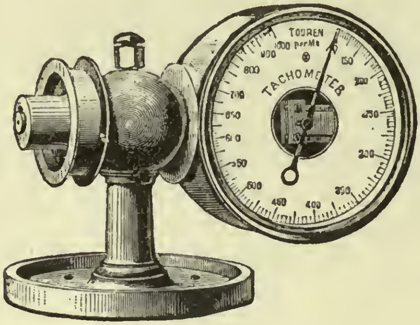


FIG. 142.—Tachometer.

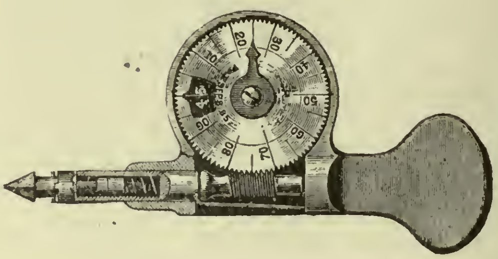


FIG. 143.—Speed Counter.

YOUNG'S SPEED INDICATOR

Fig. 144 represents an apparatus by which the speed of any shaft, of which the end is accessible, can be at once noted without timing. On the little point being pressed into the hollow at the end of the shaft the hand at once points to the number of revolutions per minute.

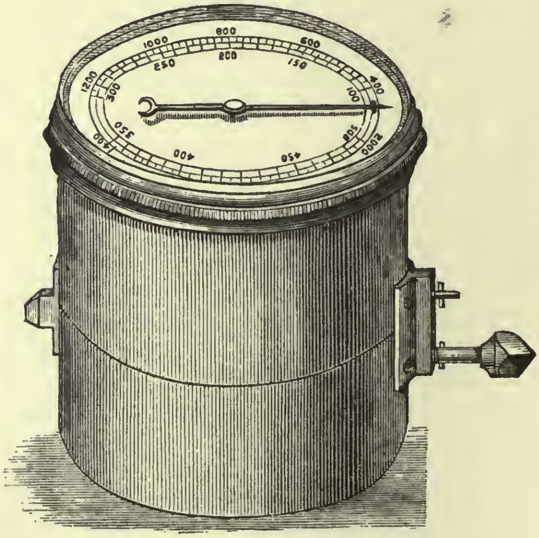


FIG. 144.—Young's Speed Indicator.

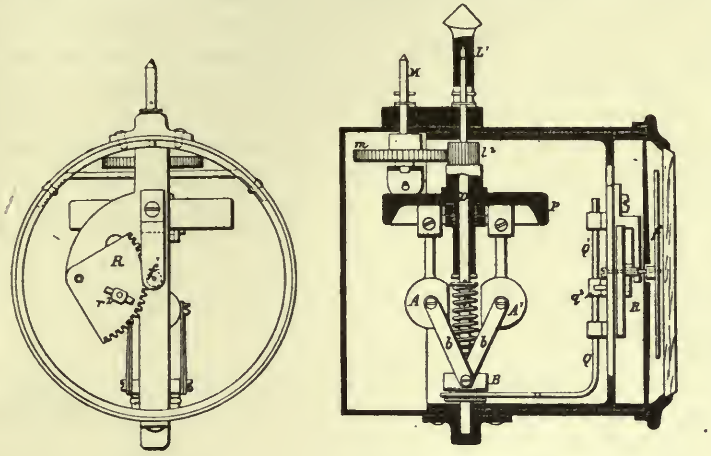
The apparatus consists of a miniature governor like an engine governor, but controlled by a spring instead of by a weight. As the balls fly out they move a lever, which moves the hand.

The point can be put on either of the two spindles shown, one being used for speeds under 500 revolutions per minute, and indicating on the inner circle on the dial; and the other for speeds up to 2000 revolutions, and indicating on the outer circle.

In Fig. 145, A, A^1 are the governor balls, which are provided with arms connected by screws to links *b b* coupled to a sliding sleeve B. The arms are coupled to the fly-wheel as shown. The fly-wheel P is fitted upon the spindle D in such a manner that when the said spindle is rotated the fly-wheel will be carried round with it by friction as soon as the inertia of the fly-wheel is overcome. The spindle D passes through the casing to receive the carrier L^1. The pinion l^2 is fixed upon the spindle D, and gears with the spur-wheel *m* fixed on the spindle M, which passes through the casing to receive the said carrier.

The sleeve or sliding bush B is constructed with a circular groove, into which fit two arms of the rod Q, the said rod being carried in lugs on the frame Q^1, and being free to slide therein. R is a toothed segment pivoted upon the frame Q^1, and provided with a slot through which passes the pin r^2, the said pin being capable of adjustment in the said slot, and being secured by means of a nut upon one end of it. The pin r^2 at its other end fits into a slotted arm q^2 fixed on the rod Q. By means of this arrangement motion is imparted by the sleeve or sliding bush B to the segment R, which actuates the index or pointer *f* by means of the pinion f^1; this gears with the toothed segment R, and is fixed on the spindle of the index.

This instrument operates as follows:—The carrier being placed upon either of the spindles, D or M is thrust against the centre of one end of the shaft whose speed is to be indicated, and the governor is thereby caused to rotate owing to the friction between the spindle D and the fly-wheel P; the balls A, A^1, flying outward by reason of the centrifugal force, draw the sleeve or sliding bush B towards the fly-wheel P, thereby compressing the spiral spring O, and at the same time acting upon the rod Q to cause it to move in the bearings. By this movement the slotted arm q^2 is caused to act upon the pin r^2, and thus turn the toothed segment R upon its pivot; the tooth segment acts upon the pinion f^1, and thereby moves the index or pointer *f*, so that it indicates upon the outer figures of the dial the number of revolutions per minute of the governor, and (if the carrier is upon the spindle D, as shown) of the shaft against which such carrier is thrust. If the carrier is placed upon the spindle M, the speed of the shaft against which it is thrust will be indicated upon the inner figures of the dial.



Plan. Sectional Elevation.

FIG. 145.—Young's Speed Indicator.

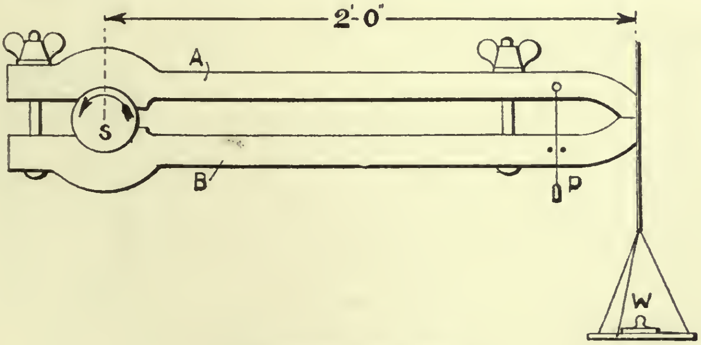


FIG. 146.—Prony Brake.

A very good tachometer may be made up of a small magneto-electric machine and a voltmeter; the voltmeter, reading proportionally to the speed, may be graduated in revolutions per minute.

After all this has been done we have only got the I.H.P. and a clear view of the action of the working fluid in the cylinder. What we want now is the brake horse-power (B.H.P.), the actual useful power given off by the engine. There is only one way to measure this power, and that is by some machine which will absorb the power and measure it.

The Prony brake is one such machine. It is shown in Fig. 146. It consists of a wooden clamp to grip the pulley or fly-wheel of the engine under tests. The lever of the clamp is weighed by weights on a scale pan hung from a vertical bar on the end of the lever. The vertical bar is intended to increase the leverage when the lever rises from the horizontal. It is usual to have two stops, between which the lever is limited in its movement. The brake is screwed up until the friction is just sufficient to support the weight. If L is the length of the lever, n the revolutions of the pulley per minute, and W the weight supported, then the horse-power $= \frac{\pi \times 2 L \times W \times n}{33000}$, or horse-power $= \frac{LWn}{5252}$ very nearly.

The brake must be designed to have a surface contact with the pulley proportional to the speed and power. If the surface is too small the friction is irregular. Again, if the surfaces are too large the brake acts somewhat indifferently. Considerable weight may be added or subtracted without moving the lever.

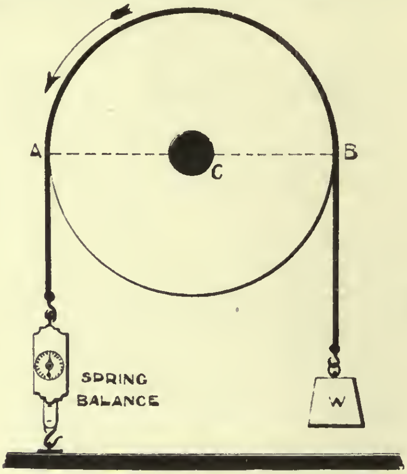


FIG. 147.—Strap Brake.

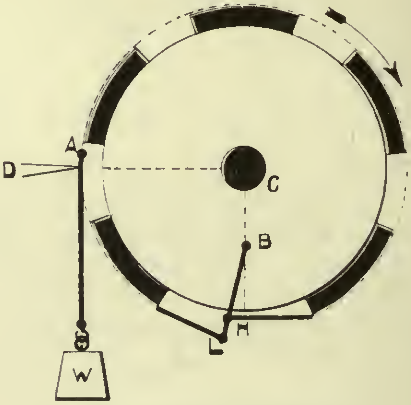


FIG. 148.—Self-Adjusting Brake.

If the surface is not lubricated the size of the brake does not make so much difference; but when lubrication is used, as it has to be in most cases, then when the surface is too small the pressure required squeezes out the lubricant, and when too large the pressure is small and lubrication irregular.

This form of Prony brake is used for smaller powers. Another form (Fig. 147) consists of a pulley on the engine to be tested, over which a strap of leather or thin brass is placed, or leather with wooden blocks. It is fastened to a spring balance on one end, and weights are hung on the other end. When the engine is running weight is added to get the necessary friction; the difference between the reading of the spring balance and the amount of weight will give the weight suspended by the friction. Suppose W = 100 lbs., and the balance indicates 20 lbs., the weight held up by friction is 100 − 20 = 80 lbs.

For larger powers it is essential that the strap should be automatically adjustable, so that the point of attachment of the weight should remain at a constant level. Fig. 148 shows the arrangement employed. The two ends of the strap are attached to two points on a lever B.H.L. suspended vertically below the centre of the brake pulley. If the friction is in excess the point A to which W is hung will be lifted above the level marked by the pointer D, and the lever B.H.L. pulled to one side. The motion of the outer end of the lever at L being greater than that at H, the strap slackens and point A descends; if the friction lessens, the point A falls and the strap tightens.

These are absorption machines destroying the power they measure. Other forms of machines are used for testing the amount of power required to drive a machine, and are called transmission brakes.

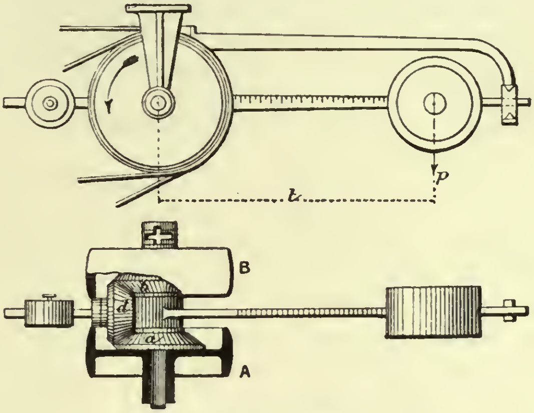


Fig. 149.—Transmission Brake.

There are many such machines, one reference, however, may suffice, known as King's dynamometer. The power is measured by a weight and by the use of bevel wheels. In it the driving and driven pulleys A and B (Fig. 149), loose upon their shaft, are connected by the bevel pinions *a*, *d*, *b*. The reactions of *a* and *b* upon *d*, each sensibly equal to the motive force exerted at the pitch circle of the pinions, are balanced by the weight W, in such a way that the turning power P of the pulley A is given approximately, that is, neglecting friction, by the same formulæ as that given for the Prony brake.

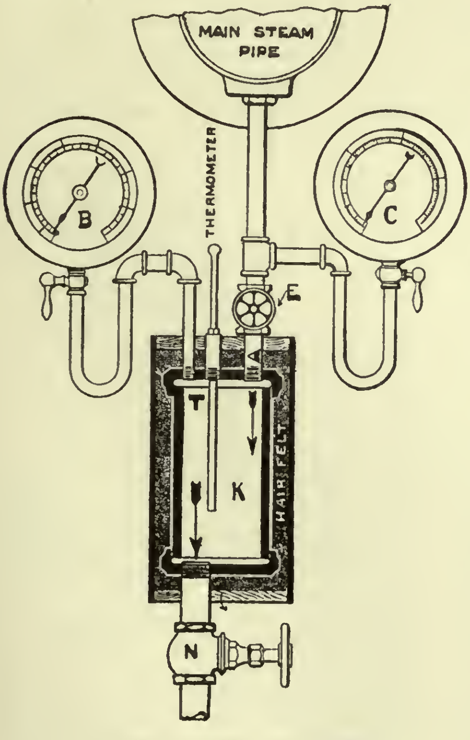


Fig. 150.—Steam Hygrometer.

A dynamo-electric machine is by far the best brake for engine tests. The power required by the dynamo itself to make up for its internal losses is easily determined, and constant at load. This power, added to the constant power indicated by an electric dynamometer or a voltmeter and ammeter, is the correct brake horse-power of the engine driving the dynamo.

Again, for testing the power required to drive a machine an electromotor is by far the best dynamometer, if the resistance of the motor, its exciting current, and other data are known, or if not they can easily be determined. Then the watts taken to drive the motor while driving the machine, minus the watts lost in the motor, at once gives the power taken by the machine under any ordinary conditions.

In testing engines it is usual to estimate the water in the steam. Water, in the shape of fine particles, cloud, or mist, is carried over by the current of steam from the boiler, and water forms in the pipes and passages to the engine from condensation, if there is no superheating. With superheated steam there is no water to estimate, for the steam is at a temperature higher than that due to the pressure. The Peabody calorimeter depends upon the principle, that steam which contains a moderate amount of water becomes superheated if the steam is expanded through a narrow nozzle from boiler pressteel piston available for steam pressures up to 280 lbs., and can be supplied with $\frac{1}{4}$ square inch area piston for higher pressures.

The cylinder is surrounded by steam, and can be easily removed for cleaning. The springs are double-wound, and made with a ball which forms a ball joint between the piston and piston rod. The movement of the piston is multiplied six times at the pencil.

The drum is ordinarily made $1\frac{13}{16}$ inch diameter by $2\frac{3}{4}$ inches high, but the indicator can also be supplied with $1\frac{3}{16}$ inch diameter drum for very high speeds up to 1000 strokes per minute.

The instrument is supplied compactly arranged in a polished oak or teak box (measuring $8\frac{3}{4}$ inches × $6\frac{3}{4}$ inches × $4\frac{3}{4}$ inches), with hinged side carrying the instrument as shown in the illustration (Fig. 154).

The complete outfit suitable for inspecting and consulting engineers at home and abroad consists of the following items:—

1. Cock with steel connection ($\frac{3}{4}$ inch Whitworth thread).
2. One bottle of oil.
3. Tap ($\frac{3}{4}$ inch Whitworth).
4. Tightening pin for loose nut.
5. Case of silver pencils.
6. Three scales.
7. Spanner for indicator body.
8. Key for screwing in spring.
9. Cylinder cleaner.
10. Screw driver.
11. One packet of metallic paper.
12. Three springs.
13. Cord adjuster.

The first item in its simpler forms is shown in Fig. 155. It has a small hole at right angles to the steam passage, whereby the water which collects in the connecting pipe is blown off. The second item consists of fine clock oil to lubricate the moving parts; the third item is often useful in tapping a connection to a cylinder for the attachment of the indicator. Indicators as a rule have a standard $\frac{3}{4}$ inch Whitworth thread. The other items are easily understood.

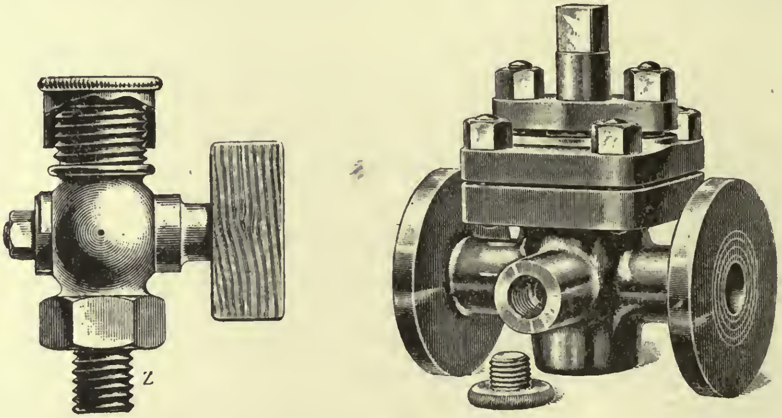


FIG. 155.—Indicator Cocks.

Most engineers fit indicator plugs to both ends of the cylinders of their engines, so that two pipes with a 3-way cock can readily be inserted when required.

For very heavy pressures and on marine engines the indicator cock made by Messrs. Schäffer & Budenberg, of extremely substantial dimensions and packed with asbestos, is shown in Fig. 155.

The same firm make Thompson's indicators to give a number of diagrams in succession on a long strip of paper, as shown in Fig. 156.

This indicator will give a series of about fifty diagrams on a continuous slip of paper, and the running mechanism is arranged within the paper drums. The paper slip, which has a length of about 6 feet, is advanced about 1 inch at every to-and-fro movement of the drum. The paper feed can also be disconnected at any time, and the instrument can then be used like an ordinary indicator.

On account of the greater moving masses of the drum these indicators can only be used for moderate speeds, namely, the large pattern up to 120 and the small pattern up to 200 revolutions per minute.

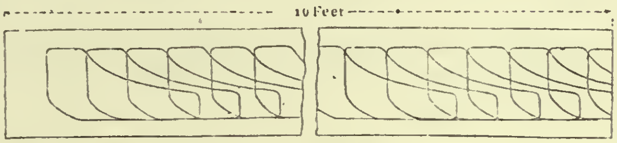


Fig. 156.—Indicator Cards.

In another arrangement the paper drum can be raised by a screw operated by hand, so that a series of cards can be drawn in a vertical column, as shown in Fig. 157. The reducing gear employed is simple and substantial, and is of a design in which both cords can be led in any desired direction, and it is therefore available in every case. The apparatus is supplied with nine small pulleys for strokes up to 16, 21, 27, 35, 49, 57, 79, 108, and 153 inches.

Two illustrations showing the construction clearly are given in Figs. 158 and 159. The wheels are fitted to a sliding sleeve on the upright standard with swivelling guide pulleys for the cords, whereby the cords may be deflected in any direction.

Having obtained the indicator cords, the next step is to find the mean pressure on the engine piston therefrom. This can be done by the method of averages as shown in Fig. 132, taking the heights of the vertical lines added together and divided by their number, the measurements being made to scale. This is a tedious, inaccurate method. The best system is to employ a mechanical averager to give the mean pressure. Amsler's planimeter is much used, also Coffin's planimeter.

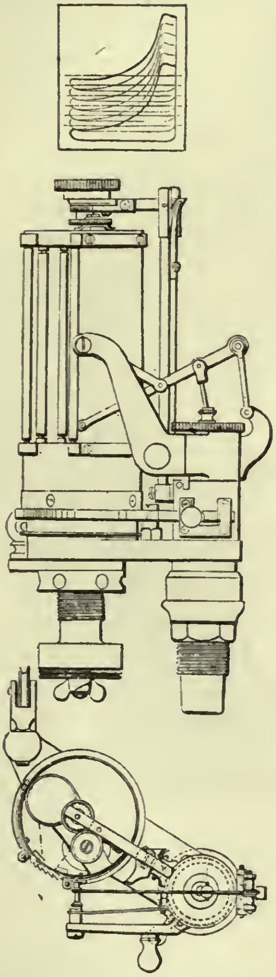
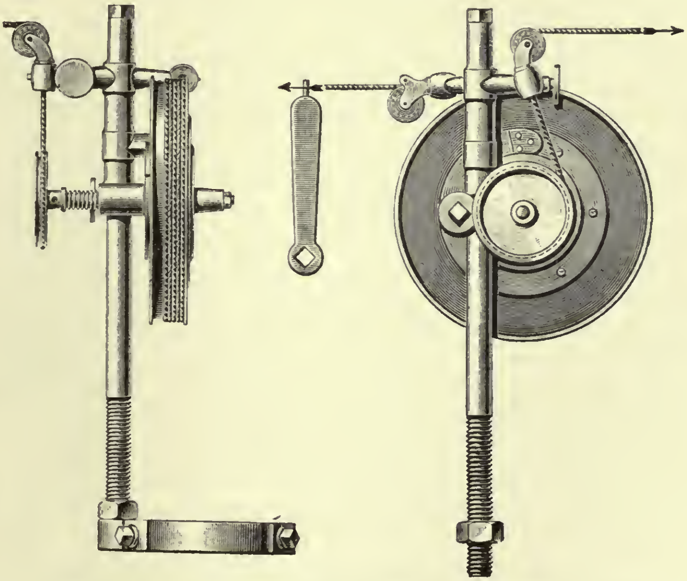


Fig. 157.—Thompson Indicator.



Figs. 158 and 159.—Thompson Indicator. Reducing Gearing.

Amsler's planimeter (Fig. 160) consists of two bars, the radius bar AB and the polar arm CD, hinged at *d* so as to have perfect freedom of motion in one plane. The needle point is kept fixed at F, while point C is carried clock-wise round the diagram. The steel piston available for steam pressures up to 280 lbs., and can be supplied with $\frac{1}{4}$ square inch area piston for higher pressures.

The cylinder is surrounded by steam, and can be easily removed for cleaning. The springs are double-wound, and made with a ball which forms a ball joint between the piston and piston rod. The movement of the piston is multiplied six times at the pencil.

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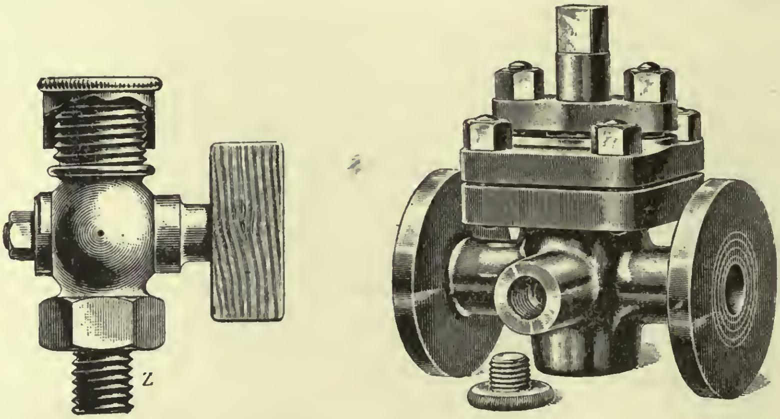

Fig. 155.—Indicator Cocks.

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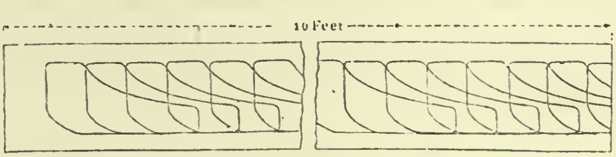

FIG. 156.—Indicator Cards.

In another arrangement the paper drum can be raised by a screw operated by hand, so that a series of cards can be drawn in a vertical column, as shown in Fig. 157. The reducing gear employed is simple and substantial, and is of a design in which both cords can be led in any desired direction, and it is therefore available in every case. The apparatus is supplied with nine small pulleys for strokes up to 16, 21, 27, 35, 49, 57, 79, 108, and 153 inches.

Two illustrations showing the construction clearly are given in Figs. 158 and 159. The wheels are fitted to a sliding sleeve on the upright standard with swivelling guide pulleys for the cords, whereby the cords may be deflected in any direction.

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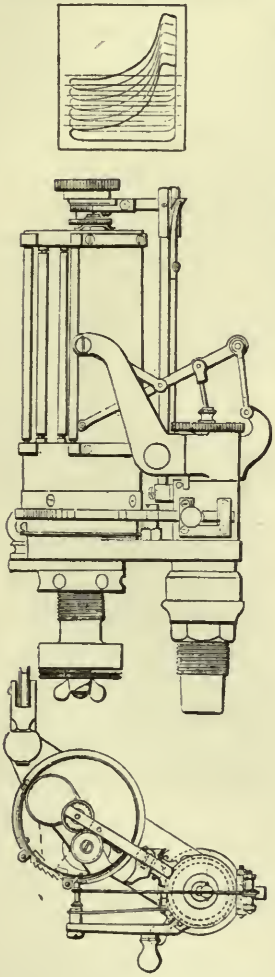
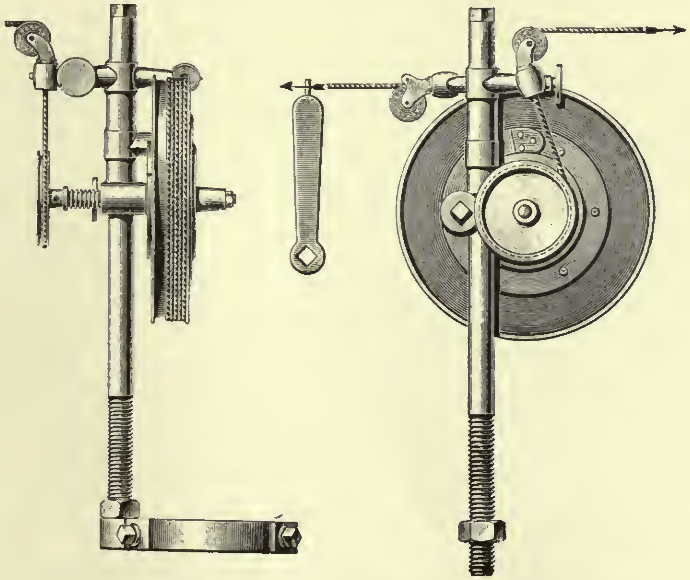


FIG. 157.—Thompson Indicator.



FIGS. 158 and 159.—Thompson Indicator. Reducing Gearing.

Amsler's planimeter (Fig. 160) consists of two bars, the radius bar AB and the polar arm CD, hinged at *d* so as to have perfect freedom of motion in one plane. The needle point is kept fixed at F, while point C is carried clock-wise round the diagram. The roller wheel P fixed to the pole arm sometimes rolls and sometimes slips; it slips without rolling when the tracer moves exactly in the direction of the axis of the arm. A scale and vernier on the roller registers the area of the diagram.

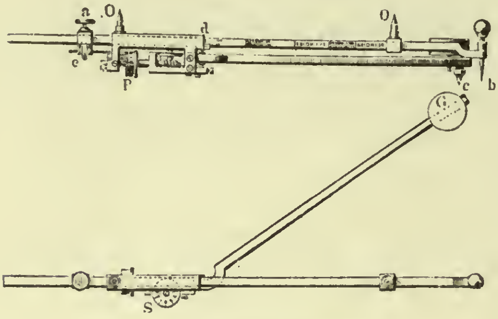


FIG. 160.—Amsler's Planimeter.

To use this planimeter the diagram is carefully pinned to a perfectly flat board. The points oo of the planimeter are adjusted so that the distance between them is equal to the length of the diagram projected on to the atmospheric line. The instrument is then placed upon the board, the point *b* being brought into agreement with any fixed point of the diagram, while the weighted point *c* is placed in a convenient position outside the diagram. The point *b* is then passed once round along the lines of the diagram. The indication on the wheel P and disc S when the point *b* has again reached the starting-point, divided by 40, gives the mean height of the diagram in inches. Supposing, for instance, that the wheel P was adjusted to stand at zero before using the instrument, and that, after tracing the diagram, the disc S—which advances by one figure for every 10 revolutions of the wheel P—stands between 1 and 2 while the wheel P indicates 21.2, a vernier being provided for reading the last figure, then the resulting number is 121.2, and this divided by 40 gives 3.03, which represents the mean height of the diagram in inches. Now, supposing that the diagram was taken by means of a No. 7 Richards' indicator spring with a scale of 32 lbs. to the inch, then the mean pressure amounts to 3.03 × 32 = 96.96 lbs. per square inch.

In practice the calculation is somewhat simplified, as the springs used are mostly of such scales that, instead of dividing by 40 and multiplying by the vertical scale, the mean pressure may be obtained by simply multiplying by a factor corresponding with the scales used.

In order to secure accurate results the instrument must be carefully cleaned before being used, and the board must be perfectly flat.

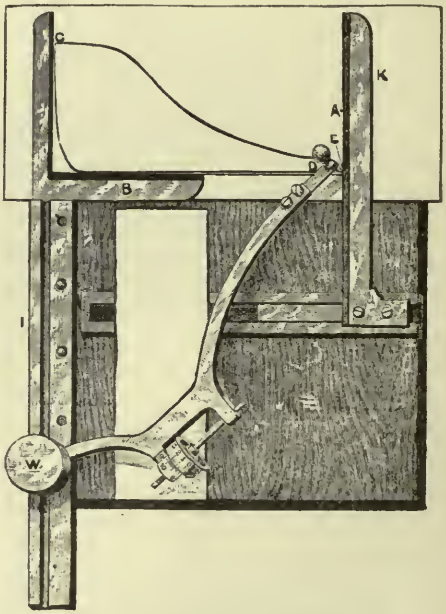


FIG. 161.—Coffin's Planimeter.

The Coffin instrument of the Globe Engineering Company of Manchester is an instrument of much the same principle, and is shown in Fig. 161. It consists of a bar carrying a registering wheel with a tracing point at one end, and a pin moving in a slot at the other end. The only moving parts are the bar, wheel, and the vertical sliding piece K; the diagram to be measured is placed as shown, the atmospheric line parallel with the edge of the square B. The vertical edges CK should almost touch the ends of the diagram. The weight W keeps the pin in the slot. The registering wheel should run on smooth flat paper.

Having placed the diagram in position, make an indentation at its extreme end, as at E, with the tracing point D, at the same time setting the wheel to zero. Then trace out the diagram by moving in a clockwise direction, until returning to the point E.

The area of the diagram may now be read off in square inches upon the wheel and vernier, but we require the mean height of the diagram. To find this we move the pointer D upwards along the inner edge of K until the wheel returns to zero. This vertical distance is the mean height to be measured by the scale of pressure corresponding to the spring used.

Goodman's averager is shown in Fig. 162, and is supplied by Messrs. Jackson Brothers, Leeds. The tracing point is fixed to the horizontal bar. The other leg of the instrument is carefully ground to a knife edge in such a way that the edge, if produced, would pass through the tracing point. The distance between the knife edge and the tracing point is adjustable by sliding the former along the horizontal bar, by means of which the instrument is set to the exact length of the diagram to be measured. Suppose the diagram, Fig. 163, is to be measured. Choose a point A somewhere about the centre of the figure, and draw any line AB to meet the boundary. The tracing point is placed upon A, and the hatchet end put so that the mean position of the instrument is roughly square with AB, as shown at X. The position of X is marked by pressing the knife edge upon the paper. Now take the tracing point lightly in the fingers, so that the movement of the hatchet end is not controlled by the pressure of the hand. Travel from A to B, then follow the direction of the arrows round the boundary of the figure, say in a clockwise direction, returning to the point B, thence back to A. The knife edge will then occupy a new position at Y, which is marked as before. The indicator diagram should now be turned about the point A through 180 degrees. The figure is again traced by the point, but this time in a contra-clockwise direction, following the dotted outline (Fig. 163). When the point returns again to A the hatchet will have travelled back towards X, say to X_1. The mean side movement of the hatchet end gives at once the average height of the diagram. Thus, suppose the distance from Y to the dot between X and X_1 is measured off as 0.56 inch, then this is the average height of the diagram, and multiplying 0.56 by the scale of pressures gives the mean pressure upon the piston. In Professor Goodman's hatchet planimeter the long calculations required to make out the average are avoided by a scale marked upon them, which enables the area in square inches to be measured by reading off the side travel of the hatchet. The instrument is so arranged that the area of diagram can be found by multiplying the length of the diagram by the side travel of the hatchet. It is a valuable instrument for quick work, not liable to damage, and is of course very much cheaper than the other instruments.

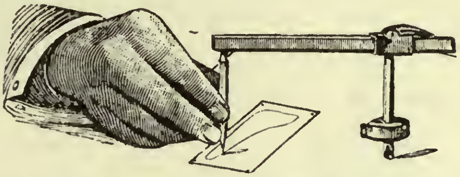


FIG. 162.—Professor Goodman's Averager.

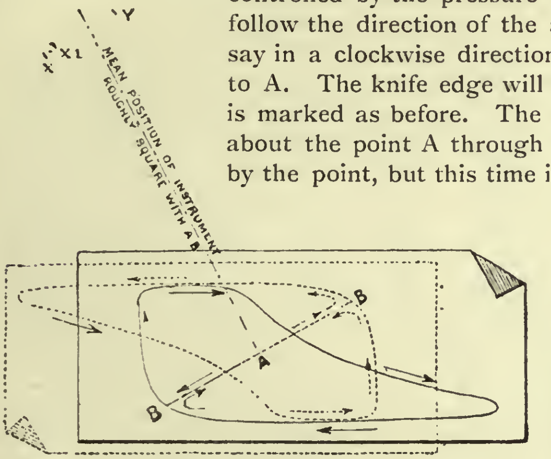


FIG. 163.—Applying Goodman's Averager.

PRESSURE AND VACUUM GAUGES

The barometer invented by Torricelli was the first scientific pressure gauge, and indicated the pressure of the atmosphere. Atmospheric pressure is of importance to the engineer, especially in these days of high vacuum.

The atmospheric pressure varies from many causes between 27 and 30 inches of mercury,—that is, it will balance a column of mercury of that height at sea-level. The pressure decreases, however, as we ascend above the sea-level. From the height of the column of mercury we can calculate the pressure per square inch. For if we assume the column to have a cross section of 1 square inch, and the weight of a cubic inch of mercury to be roughly estimated at $\frac{1}{2}$ lb., the pressure of 30 inches will be equal to 15 lbs. on the square inch. This pressure is called one atmosphere more accurately in the metric system. One cubic centimetre of mercury at zero weighs 13.596 grammes. The pressure of 760 millimetres $= 13.596 \times 76 = 1033.3$ grammes per square centimetre = 14.7 lbs. per square inch. Barometers may be found described fully elsewhere.

The mercury column vacuum gauge (Fig. 164) consists of an open glass tube dipping into a reservoir of mercury. The vacuum space to be measured is connected to the top end by a flexible tube. The height to which the mercury rises can easily be measured by a scale alongside of the column, or by an ordinary foot rule.

Vacuum gauges are now usually made the same as pressure gauges to be presently described, on the Bourdon or corrugated diaphragm system.

A mercurial pressure gauge (Fig. 165) is sometimes used for checking the actual pressure indicated by pressure gauges. It consists of a strong reservoir *a* containing mercury. Into this reservoir a glass tube dips. By means of a force pump water can be pressed in above the mercury, and this forces up the column in the tube. The height of the column can be measured on a scale, and from that the exact pressure calculated. To check a gauge it is only necessary to couple it to the water space above the mercury. Its scale reading can then be compared with the pressure shown by the mercury column's height.

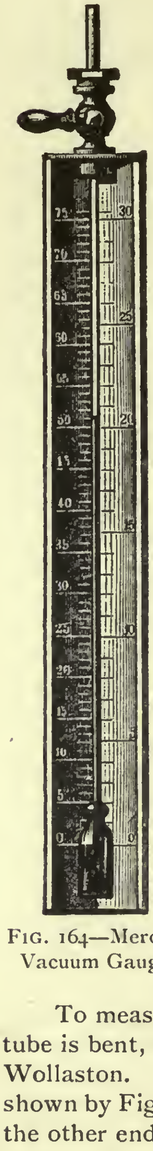


FIG. 164—Mercury Vacuum Gauge.

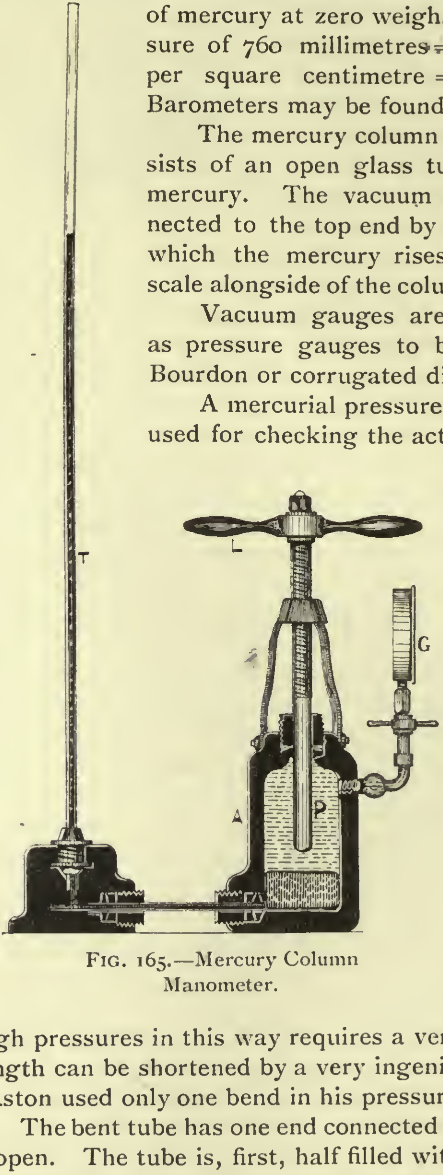


FIG. 165.—Mercury Column Manometer.

To measure high pressures in this way requires a very long tube. If, however, the tube is bent, the length can be shortened by a very ingenious invention, first made by Dr. Wollaston. Wollaston used only one bend in his pressure gauge. The principle may be shown by Fig. 166. The bent tube has one end connected to the pressure to be measured; the other end A is open. The tube is, first, half filled with water, and then the tube and vessels are filled up with oil to about one-fourth the depth of the vessels. The specific gravity of the oil should be 0.9. When the pressure is turned on the water sinks in BE and rises in AD, until the levels are PQ and DE. Here we have opposed to a column of water PD a column of oil QE.

Now, if S be the sectional area of the tube, then the (pressure in B − the pressure in A) S = equals the weight of $PD - QE = \frac{1}{10}$ the weight of PD. Hence PD must be 10 inches to balance a difference of pressure in B equal to 1 inch. This gauge is only applicable for small pressures. For higher pressures a multiple tube arrangement is made, and the two fluids are mercury and water. The apparatus for this purpose consists of an iron tube ABCD bent back upon itself a certain number of times (Fig. 167).

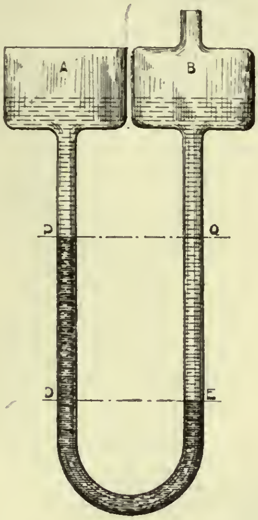


Fig. 166.—Differential Manometer.

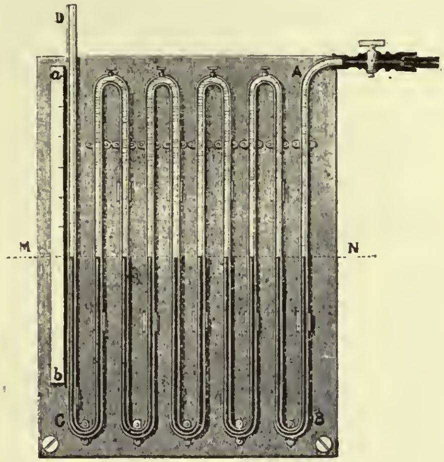


Fig. 167.—Multiple Differential Manometer.

The extremity A communicates with the boiler by a stop cock, and the last branch CD is of glass, and has a scale by its side.

The graduation of this scale is effected directly in practice by placing the manometer in communication with a reservoir of compressed air, whose pressure is given by an open mercurial gauge, or by a standard manometer of any kind.

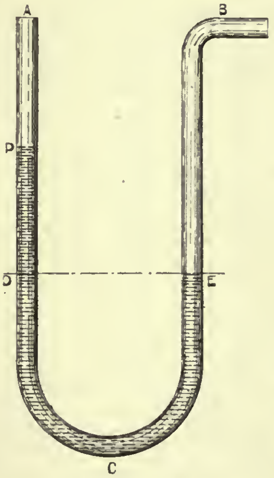


Fig. 168.—Simple Mercury Manometer.

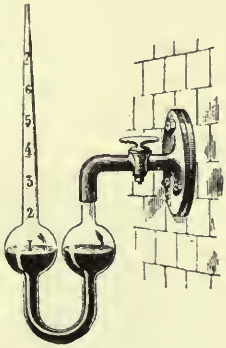


Fig. 169.—Mercury Gauge.

If the tube AB be supposed cylindrical, the graduation can be previously effected by an application of Boyle's law (Fig. 168).

The sensibility of the instrument decreases as the pressure increases, and the distance traversed by the mercury for an increment of pressure equal to one atmosphere becomes less and less. This inconvenience is partly avoided by the arrangement shown in Fig. 169. The branch containing the air is of a conical form. In this way, as the mercury rises, equal changes of volume correspond to increasing lengths. The effect of this arrangement is seen by an inspection of the scale, on which the numbers corresponding to successive atmospheres of pressure are nearly equidistant, whereas when the tube is cylindrical they rapidly approach each other.

This tapering gauge tube has been successfully used for measuring the high vacuum of electric incandescent lamp bulbs on the exhaust pump. By forcing the contents of a large bulb into a tapered tube the gases are compressed and measured.

These fluid pressure gauges are not portable. They are generally fixed as permanent gauges for accurate comparisons. Portable gauge testers have been introduced for the rapid testing of gauges even when under steam on the boiler. They are built on the principle of the hydraulic ram, and dead weights of known value on a piston of known area give the desired pressure directly.

The principle is shown in diagram, Fig. 170. The piston S carries the dead weight W. Water is filled in below the piston at F through the screw plug *a*. This plug has a plunger P, by means of which the water is compressed until it raises the weight W. Now, if the plunger has $\frac{1}{10}$ inch area, and the weight lifted is 10 lbs., the pressure is evidently 100 lbs. per square inch.

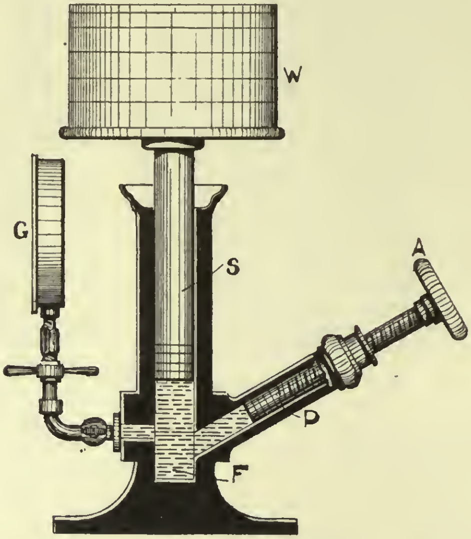


FIG. 170.—Gauge Tester.

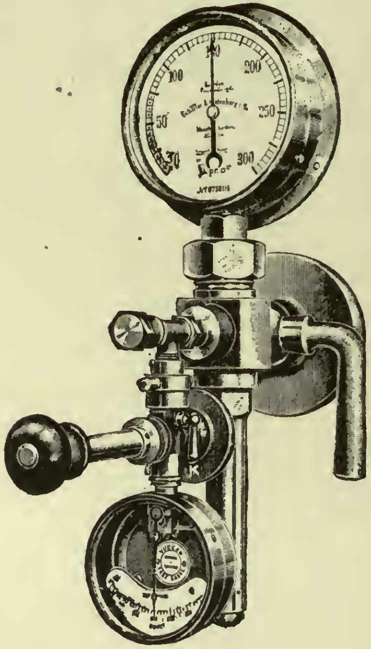


FIG. 171.—Gauge Tester.

These testing machines have been brought to great perfection by Messrs. Schäffer & Budenberg. One of their type is shown (Fig. 171) testing a gauge on a boiler. In this case, however, instead of the dead weight there is a standard pressure gauge showing the correct pressure.

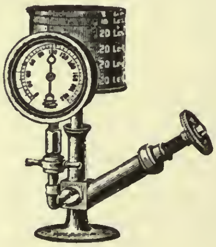


FIG. 172.

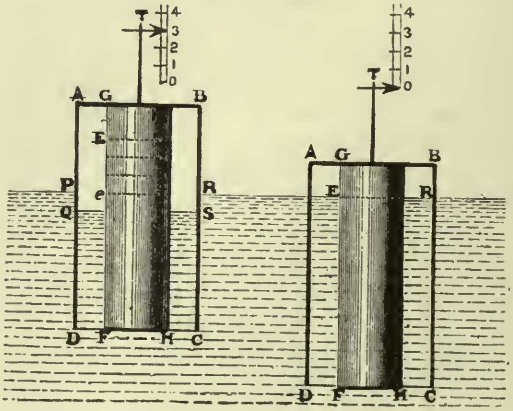


FIG. 173.—Sensitive Multiplying Manometer.

The small figure (Fig. 172) shows the machine with dead weights on the scale.

A pressure gauge acting on the fluid principle, highly sensitive for small pressures such as are used in gas distribution, can be constructed as follows (Fig. 173). A hollow float is attached to a bell gas holder, which is connected by pipe to the main gas pipe, the pressure of which is to be measured.

The volume of the float to be one-third of the capacity volume of the bell. If the difference of the level of water inside and outside is made equal to PQ, or 1 inch, the gas will be under a pressure of 1 inch, but the holder or bell will rise 3 inches. If the sectional area of the float is $= a$, and the sectional area of the annular space $= 3a$ and $W =$ the weight of a cubic inch of water, then the volume of water displaced by the pressure in $PQRS = 3a \times PQ$. The lifting force $= 3aPQ \times W$.

The weight on the float is lessened by that due to the pressure on the upper area of the bell, hence the float will rise to a height Ee, such that $Ee \times a \times W = 3a \times PQ \times W$. Therefore $Ee = 3PQ$, or the pointer rises 3 inches for 1 inch displacement, and thus a very sensitive low-pressure gauge is obtained.

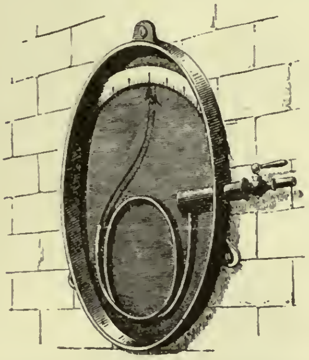


FIG. 174.—Original Bourdon.

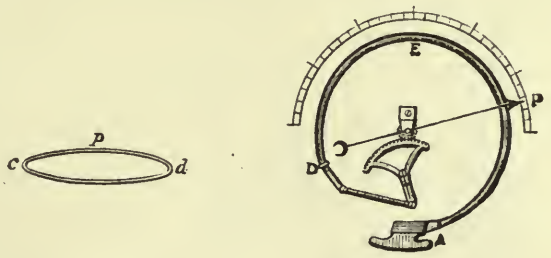


FIG. 175. FIG. 176.

Bourdon Gauge.

For most engineering purposes the ordinary Bourdon gauge and disc gauges are sufficient. The Bourdon has long held a foremost place. It originally was a very simple machine, without any gearing. It consisted simply of a flat metallic tube coiled up into about two turns with a pointer on the free end and the other end fixed to the case, and a tap as shown in Fig. 174. It depends upon the principle that a flat bent tube tends to straighten out when there is a difference of pressure between the inside and outside, and the difference of pressure greatest on the inside of the tube. If the greater pressure is on the outside of the tube it bends in farther, and so can also be used as a vacuum gauge. As ordinarily constructed it now consists of a comparatively short tube, and the motion is multiplied by a rack and pinion and a lever and link motion, as shown in Fig. 176. The section of the tube is shown in Fig. 175.

In order to protect the gauge from the heat of steam it is connected through a syphon pipe filled with water. The steam is therefore not able to enter the gauge, but its pressure is transmitted by the column of water in the syphon. The flat tubes are usually of brass, different thicknesses of walls being used for different pressures.

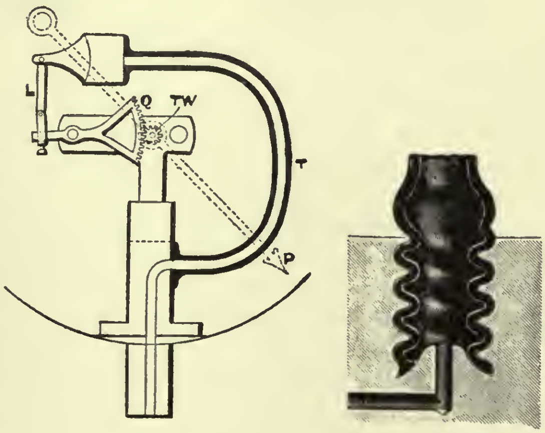


FIG. 177. FIG. 178.

High-Pressure Bourdon.

For very heavy pressures steel tubes are used, shaped as shown in Fig. 177, and special connections have to be made between the tube and its fixing to make them secure under the high pressures. This necessity for a specially safe joint has been met in the Schäffer & Budenberg gauges by corrugated joints solidly secured in the metal inlet block, and clearly shown in Fig. 178.

According to Admiralty specification, "the spring tubes of the gauges of and over 500 lbs. per square inch are to be screwed as well as soldered into the shoes and lifters." This metallic joint, shown in Fig. 178, is accepted by the Admiralty in lieu of this arrangement. This joint is absolutely tight and safe under all pressures without reliance on soldering, and cannot be in any way affected by the highest pressures. In fact, the high pressure tends to expand the metal of the tube all the tighter into its socket.

The other type of gauge (Fig. 179) has a corrugated disc with concentric corrugations. This disc springs up under the pressure. The disc is of steel, and held firmly by the flanges all round its edge. The centre of the disc is coupled by a rod to the rack quadrant. The motion of the disc is proportioned to the difference of pressure between its sides.

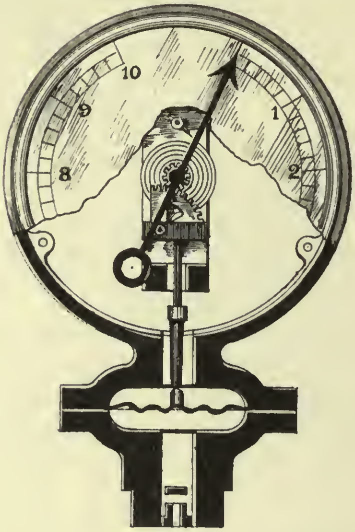


FIG. 179.—Diaphragm Pressure Gauge.

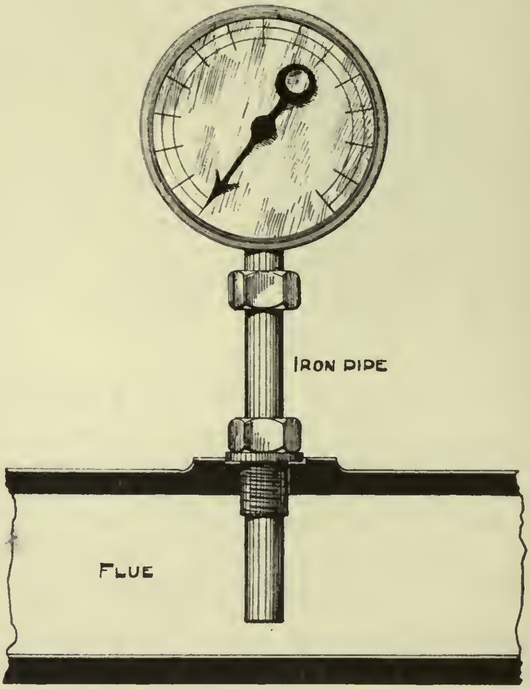


FIG. 180.—Air Thermometer.

In addition to these gauges, thermometers are used to obtain the temperatures of flue gases and of furnaces. For flue gas temperature tests an air tube fitted to a pressing gauge is used, under different names—pyrometers, thalpotasimeters, and so on. They are all thermometers indicating the temperature, only some are designed for low temperatures and others for high temperatures.

The mercury thermometer is quite reliable between a few degrees above its mercury freezing-point, the zero of Fahrenheit scale, and about 500°. Between 500° and 1000° some form of air thermometer is satisfactory with careful handling, but beyond that temperature it must be admitted that the thermometers hitherto devised have considerable drawbacks.

The air thermometer for testing flue gas temperatures is shown in the diagram, Fig. 180. The pressure gauge is a sensitive Bourdon or disc type of gauge. The pipe which dips into the gases is calculated to contain a certain volume of air. The expansion of this air due to the temperature causes a pressure proportional to the temperature. The instrument really measures air pressure, but that being proportional to the temperature, the scale is graduated in degrees Fahrenheit instead of in pounds per square inch.

For still higher temperatures than those reached by the air thermometers the Siemen's electrical thermometer (Fig. 181) is perhaps the best yet devised. It depends on the electrical resistance of metals increasing with increase of temperature. A platinum spiral of fine wire .01 inch diameter, the resistance of which averages 3.6 ohms per yard of length. This wire is coiled upon a cylinder of hard baked pipe-clay in which a double threaded helical groove is formed, to prevent the convolutions from coming into contact with each other. The form of pipe-clay cylinder is shown in Fig. 183. At each end of the spiral portion BB it is provided with a ring-formed projecting rim *c* and *c*′, the purpose of which is to keep the cylinder in place when it is inserted in the outer metal case, and to prevent the possibility of contact between the case and the platinum wire. Through the lower ring *c*′ are two small holes *b b*′, and through the upper portion two others *a a*′. The purpose of the upper holes *a a*′ is for passing the ends of the platinum wires through before connecting them with the leading wires. From these two holes, downwards, platinum wires are coiled in parallel convolutions round the cylinder to the wire, and providing a connecting screw link of platinum L, by which any portion of the loop can be cut off from the electric circuit.

The pipe-clay cylinder is inserted in the lower portion AA of the protecting case, shown in Fig. 182. This part of the case is made of iron or platinum, and is fitted into the long tube CC, which is of wrought iron, and which serves as a handle. When the lower end of the casing is of iron there is a platinum shield to protect the coil on the pipe-clay cylinder. The purpose of the platinum casing is to shield the resistance wire against hot gases and against accident. At the points AA (Fig. 183) the thick platinum wires are joined to copper connections, over which pieces of ordinary clay tobacco-pipe tube are drawn, and which terminate in binding screws fitted to a block of pipe-clay closing the end of the tube. A third binding screw is provided, which is likewise connected with one of the two copper connecting wires, and which serves to eliminate disturbing resistances in the leading wires.

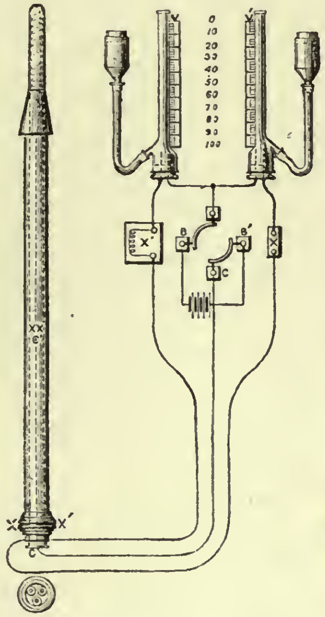


FIG. 181.

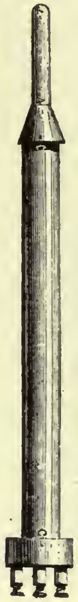


FIG. 182.

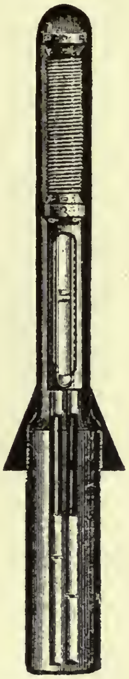


FIG. 183.

Siemen's Pyrometer. Electric.

If temperatures not exceeding a bright red heat are to be measured the platinum protecting tube may be dispensed with, and iron or copper substituted.

The pipe-clay tube was found to be highly insulating when cold; when heated, its conducting power increases, though not to such an extent as to occasion any perceptible error.

In adapting the pyrometer to the measurement of high temperatures a wide range of resistance is obtained, and it is no longer necessary to determine these resistances with the same precision as in measuring slight variations of ordinary temperature; and the use of galvanometers employed in the electrical thermometers already referred to is dispensed with, for which is substituted the differential voltameter. We need not here go into the electrical science of this part of the instrument. Suffice it to say that it is an accurate and delicate instrument for ascertaining the electric resistance of the platinum wire coil. Other methods of ascertaining the value of this resistance would nowadays be employed.

Another thermal measuring apparatus required by the engineer is a calorimeter, an instrument to ascertain the amount of heat given out by 1 lb. of fuel burning with the oxygen of the air.

The chemist Welter propounded his law that all combustible bodies disengage the same amount of heat during their combination with the same weight of oxygen; or, in other words, that the heat developed was proportionate to the amount of oxygen assimilated during the combustion. Welter founded this theory upon the results of Laplace, Lavoisier, Despretz, Rumford, and others.

Berthier has founded a system of analysis upon this theory, by which the oxygen is estimated, and therefrom the heating power of the fuel deduced. It is easy and expeditious, and may, in the case of pure carbon, or of fuel consisting of carbon without admixture of other reducing agents, be employed with advantage, and will be found to give correct results; if hydrogen is present, however, in the fuel, the result obtained would be erroneous. This appears from the consideration that one part by weight of hydrogen will reduce the same amount of oxide of lead as three parts of carbon; that is, hydrogen combines with exactly three times the quantity of oxygen that carbon takes up, and so from Welter's theory the heat developed should be three times greater when hydrogen is burned than when an equal quantity of carbon is consumed. The process is not applicable to the determination of the calorific powers of fuels composed of variable quantities of hydrogen and carbon.

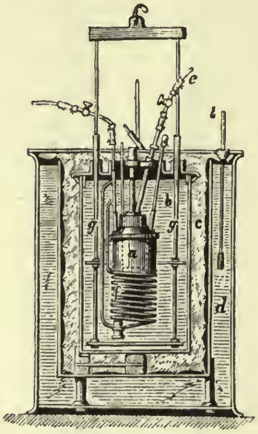


Fig. 184.—Silbermann Calorimeter.

The determination of calorific values of fuels was fairly accurately made by the experiments of Favre and Silbermann, and were performed in the apparatus shown in Fig. 184, by means of which a large series of experiments on the development of heat was made on a larger scale even than those of Andrews, already referred to, the experimental results in the two cases being generally found to agree. Favre and Silbermann's apparatus consisted essentially of a vessel of brass gilt *a*, in which the combustion took place; this vessel was placed in the calorimeter *b* of silvered copper containing water. The calorimeter was supported in another vessel *c*, lined with swan's down, and was itself surrounded by a double cylinder *d*, the annular space of which was filled with water. By this means the external influence of the atmosphere upon the internal temperature was found to be reduced to a minimum, and was easily measurable. When the combustions were performed in oxygen the gas was first dried and then allowed to flow into the combustion chamber by the tube *e*, and the gases produced and the superfluous oxygen were made to traverse a spiral tube of thin copper *f*, so as to assume the temperature of the water before quitting the apparatus. Uniformity of temperature was produced by means of the agitator *g*, *g*. The apparatus shown in Fig. 184 is the arrangement for burning carbon. Solid bodies were kindled by the introduction of live charcoal, liquids were burned in lamps having asbestos wicks, and gases were introduced by a jet previously inflamed. The weight of substance burnt was ascertained by weighing the products of combustion. The scale of the thermometers allowed of a reading of .01 of a degree C.

Another calorimeter, known as the bomb calorimeter, is made by Messrs. Bryan Donkin & Co., London.

It consists essentially of a strong metallic shell of bronze, gilt, or enamelled inside, closed by a screw-down lid. The fuel to be tested is burned inside on a spoon, and ignited by a fine wire raised to a white heat by electricity. Before ignition oxygen is pumped into the shell to a pressure of 300 lbs. per square inch, so that the combustion is complete and instantaneous. The heat produced is calculated from the temperature rise of a water-jacket which takes up the heat of the combustion. It is perhaps the best calorimeter for liquid and solid fuels.

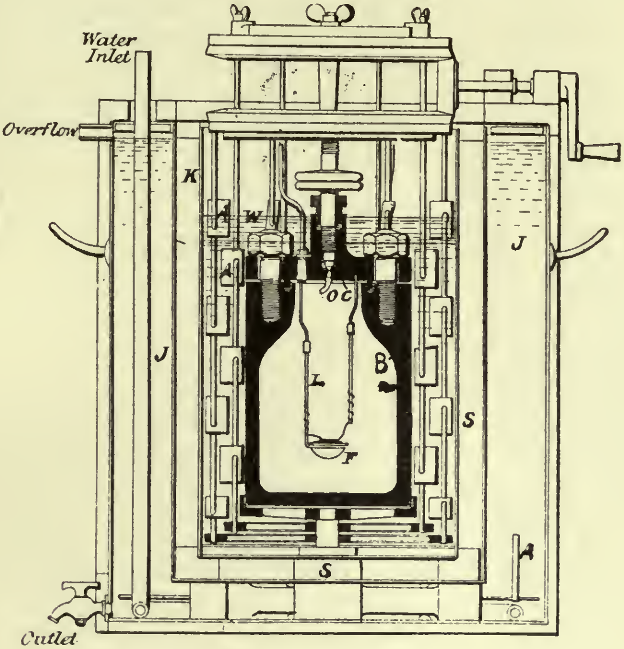


FIG. 185.—Bomb Calorimeter.

Fig. 185 is a section showing the complete arrangement; the outer jacket of water is an insulator of heat, the inner water with mixing paddles is the heat absorbent to be measured for temperature.

JUNKER'S GAS CALORIMETER

This instrument, supplied by Mr. Hermann Kuhne, London, is the best for gaseous measurements. The principle of the instrument is that the heat generated by a gas flame is absorbed by a water jacket. The quantity of water, its inlet and outlet temperatures, and the quantity of gas passing through the instrument, afford data from which the calorific value of the gas may be determined to within 0.5 per cent. of any other determination. Radiation is prevented by surrounding the apparatus by an air jacket formed by a nickel-plated cylinder. The flame shown (Fig. 186) is surrounded by a water jacket, through which pass a number of vertical copper tubes. The flame burns in the central chamber, and the products of combustion pass down the inside of the tubes to the outlet. By this arrangement the gases at the highest temperature meet the hottest water, whilst as the gases cool they meet the colder inlet water. This, of course, favours the transmission of heat, and the products of combustion escape at the throttle at atmospheric temperature, having parted with the heat of combustion to the jacket water. In order to obtain accurate results it is necessary that the flow of water through the instrument should be perfectly uniform. This is secured by leading the water into a tank shown (Fig. 186). This tank is kept slightly overflowing, thus producing a constant head. Two thermometers are fitted for measuring the temperatures of the inlet and outlet, and a graduated measuring glass is supplied for water measurements. It is highly important that the gas pressure remain constant, and for this reason a suitable regulator is required to be used with the calorimeter. An external view of the apparatus showing the gas meter and regulator is given in Fig. 187. Water formed by the combustion in the central chamber is collected in a small measuring glass at *d* (Fig. 187).

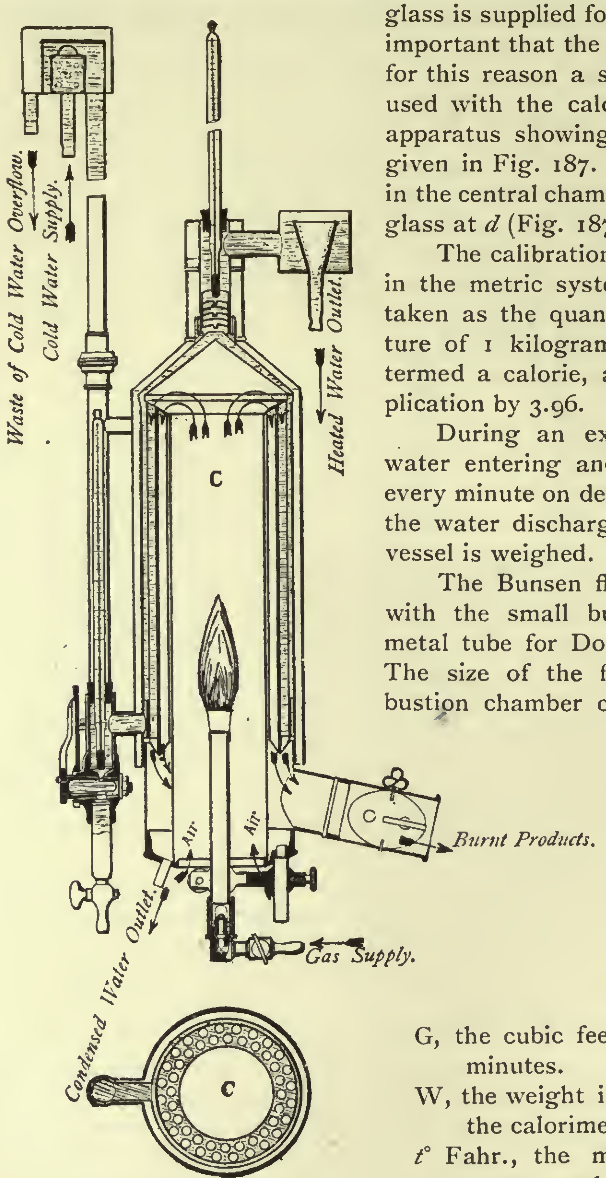


FIG. 186.—Junker's Calorimeter.

The calibrations of the instrument have been made in the metric system, and the unit of heat has been taken as the quantity required to raise the temperature of 1 kilogramme of water 1° C. This unit is termed a calorie, and is converted into $\ominus$ by multiplication by 3.96.

During an experiment the temperature of the water entering and leaving the water jacket is read every minute on delicate thermometers (Fig. 187), and the water discharged into a measuring glass or other vessel is weighed.

The Bunsen flame should be blue or colourless, with the small burner for coal gas, and the plain metal tube for Dowson gas or Mond cheap fuel gas. The size of the flame and its position in the combustion chamber can be adjusted. Care should be taken that no air from the gas pipes or connections is mixed and measured with the gas in the meter, and that there are no leaky joints between the gas meter and burner. When the flow of water is adjusted, and the temperature of the outlet water steady, measurements are made of—

G, the cubic feet of gas burned in ten or fifteen minutes.

W, the weight in pounds of water passed through the calorimeter in that time.

t° Fahr., the mean rise in temperature of the water; and

w lb., condensed water collected from the gas burned.

From these observations it is quite clear that the gross calorimetric value of the gas is equal to $\frac{Wt}{G}\ominus$ per cubic foot.

The available heat or effective heating value of the gas in the gas engine cylinder is obtained by deducting the latent heat of the steam formed per cubic foot of gas—that is, $\frac{966\,w}{G}$.

ANGULAR VELOCITY METER

In single-cylinder engines it is hardly possible to obtain constant angular velocity during one revolution of the engine, there being always a slowing down at the dead points, and an acceleration at the maximum impulse twice in each revolution. This can be detected most readily by a telephone connected to an electric generator, driven by the engine, with a transformer between the generator and telephone. The rise and fall in speed produces a regular rising and falling in the note heard in the telephone. This apparatus merely detects the irregularity, but cannot measure its amount, and perhaps this measurement could be effected with a voltmeter; but no instrument has been brought out for the purpose.

GOVERNOR TESTING

To test the efficiency with which any governor maintains an even speed with varying loads, the tachometer (Fig. 142, page 98) and the speed indicator are now much used. Every engine to be used for electrical generators should be carefully tested in this way by a speed indicator, by switching off and on the whole load, and noting the difference in speed between no load and full load, the boiler pressure being steady and the stop valve full open.

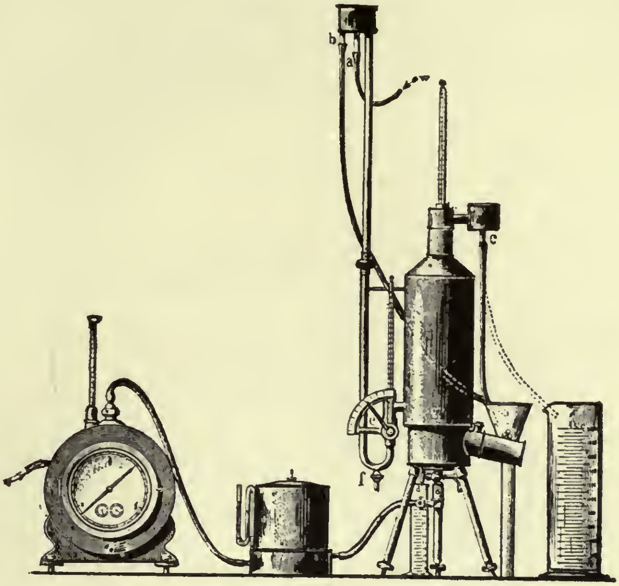


Fig. 187.—Junker's Gas Calorimeter.

TESTING STEAM CONSUMPTION

A most important test of a steam engine is that for ascertaining the steam consumption at different loads. The most correct method is to use a surface condenser to condense the steam used over an hour, a half-hour's run at each load. The condensed water collected at each test is then carefully weighed.

If the total efficiency of a complete plant is to be ascertained, then the coal and water fed into the boiler during five or six hours' run at constant known load must be carefully measured. This test shows the efficiency of the plant between the coal-pile and the work done, and is, after all, the important test for the steam user, as it lumps together all the steam used in the engine and its accessory pumps, economisers, blowers, condensers, leakages, and waste generally.

Many a high-class engine is a member of a very inefficient complete plant. High efficiency is seldom maintained equally all over a system. It is generally highest in those portions of the plant where inefficiency is most obvious and more easily measured.

ECONOMY DIAGRAM OF A STEAM ENGINE

Professor Weighton has treated the various losses from the coal-pile to the load on the engine from end to end, set out in the diagram, Fig. 188. In testing engines it is necessary to devise some means of estimating or ascertaining the losses set out herein. As an example, take a marine engine.

Taking the coal with 100 units of heat energy we lose 4 per cent. by imperfect combustion. About 20 per cent. in the chimney condensation and radiation come next. Then comes the natural loss in steam, the latent heat or heat required to maintain it as a gas, so that the steam engine could not by any possible means give a higher efficiency than 19.5 per cent. Condensation in the engine and friction brings this down to 10.67 per cent., and if applied to a screw propeller the inefficiency of the screw would bring it down to 9.26 on the screw; and if we went still further, and find the work actually done in propelling the ship forward at its ordinary rate, it would not be more than 5 per cent. of the energy expended.

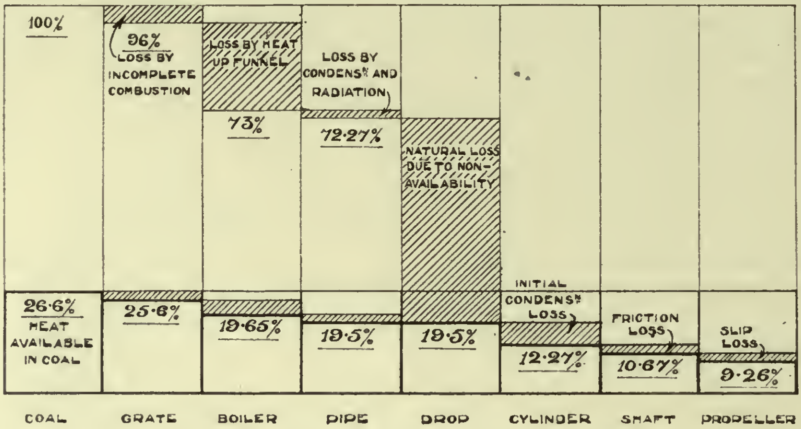


FIG. 188.—Diagram of Steam Engine Inefficiency.

THE WEISS COUNTER-CURRENT CONDENSER

This condenser, working on the syphon principle, has been introduced by Messrs. Richardsons, Westgarth, & Co. It is claimed for it that it will work with hotter condensing water than any other, and hence is most suitable for condensers using artificially cooled water.

The whole arrangement is shown in Fig. 189.

The steam enters the lower part of the condensing vessel through the exhaust pipe, which, after entering the condenser, turns downward.

The water enters the upper part of the condensing vessel, and falls in cascades, condensing the steam and running out by gravity, together with the condensed steam, through the barometric tube which forms the lower part of the condenser.

The air is drawn out at the top of the condenser, passing through the separator on its way to the air pump.

The condenser proper can be supported on an independent tower, as shown in the illustration, or in any other convenient manner. It need not be sheltered from the weather, nor is it necessary to have the condenser in close proximity to the main engines. The lower end of the condenser is placed about 33 feet above the surface of the water in the hot well.

The air pump is a "Dry" air pump, abstracting only air from the condenser.

The design of the air cylinder is practically the same as that of a common steam cylinder, having a metallic packed piston and a slide valve of the same type as a steam valve, the latter facilitating the admission as well as the discharge of the air. This air pump is in reality an efficient type of air compressor. The air valve is of the equalising type, eliminating the losses due to clearance. It is operated by an eccentric, which gives it a positive motion, thus enabling the pump to run at a high speed.

The power for driving this air pump, and also the circulating pump, is obtained from an engine of a refined type, automatically regulated by a capacity governor.

The water pump can be of any positive action type. In the diagram is shown a rotary pump driven by the engine operating the air pump already referred to. Both the pump and the engine driving it have sufficient power to lift the water a height equal to the whole difference in levels between the supply well and the level where the pipe enters the condenser.

The pump does not require to maintain the difference of level; the vacuum relieves the pump of the pressure corresponding to the barometric column, leaving only the difference of height to be kept up by the pump.

The prevailing impression is that with widely fluctuating loads condensation is incorrect. No engine, whether condensing or non-condensing, will run as economically with a fluctuating load as with a steady one. But the relative increase in economy due to condensation is as great.

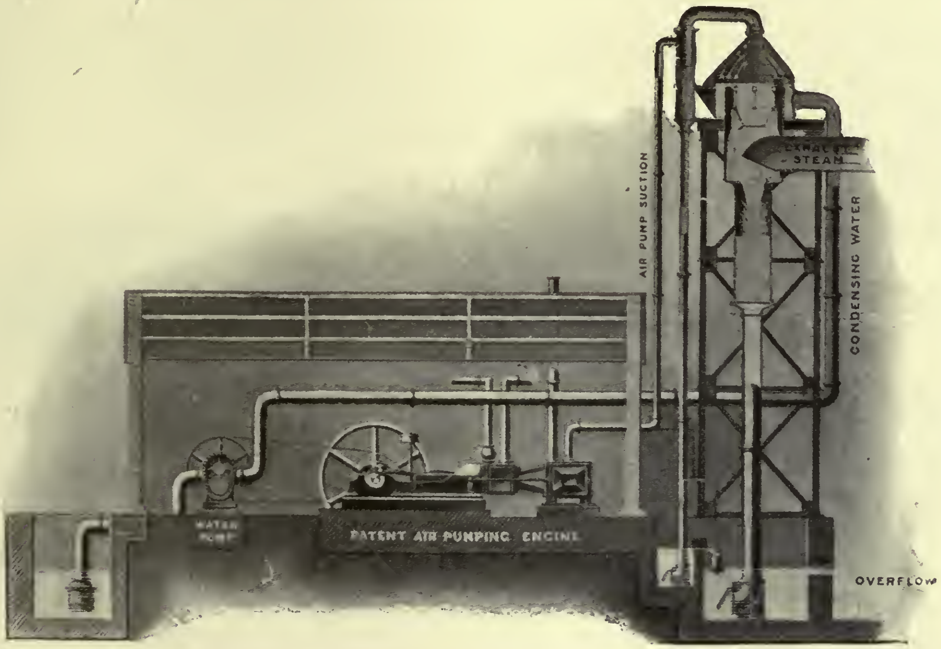


FIG. 189.—Weiss Counter-Current Condenser.

In fact, the increase in economy due to condensation is greater when an engine is much over-loaded, or when it is much under-loaded, than it is when the engine is running at its normal capacity.

Since an engine with a widely fluctuating load is most of the time either over or under-loaded, the economy of using a condenser will be correspondingly great.

An ordinary condenser has difficulty in holding its vacuum when applied to such engines as rolling-mill reversing engines, whereas the Weiss condenser has none. The false impression regarding the economy of condensation in such cases is probably due to the fact that heretofore there has been no condenser in the market capable of satisfactorily condensing the steam from such engines.

The better practice is to use a single condenser for all of the engines within reach, irrespective of the kind of engine or of the kind of load. The condenser is as independent as the boiler. The exhaust main can be run throughout the works, and all engines attached to it, just as a great variety of engines, small and large, can be attached to the same steam pipes.

Owing to the tenacity with which the Weiss condenser holds the vacuum, one condenser, one air pump, and one water pump will serve to condense the steam from many steam engines.

There are Weiss plants running successfully with the condensers placed more than 1000 feet from their engines, and the air pumps 600 or 700 feet beyond the condensers.

In using the condenser in a central condensing plant, ideal conditions exist. If one or more engines are stopped, or lose part of their loads, a higher vacuum is at once obtained in the engines that remain in service.

CHAPTER III

WATER SEPARATORS, EVAPORATORS, STEAM TRAPS, AND SUPERHEATERS

THE water separator should not be necessary in modern plants. If the boilers are correctly built and connected up, and a good drainage system fitted for blowing off water before starting, and superheating adopted as it should be, the separator is not necessary. But in many cases boilers prime, and there is no superheat. Condensation goes on all the time, and a separator is then necessary. And with the separator a good steam trap is a necessary complement; in fact, the two should be sent out for fitting up together.

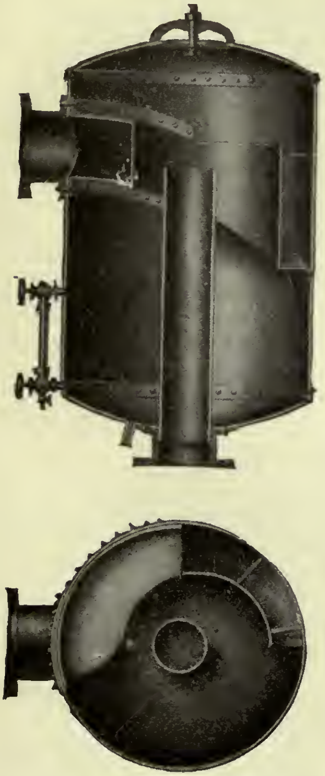


FIG. 190.—Centrifugal Separator.

The number of so called steam and water separators and traps which have been brought out is very large, yet few engineers can say that an entirely satisfactory separator or trap has been found. Recently, however, more attention has been given this subject by competent engineers, and better results are now obtainable. As an example of a separator acting on the centrifugal system, Fig. 190 is a sectional plan, showing how the fluid enters at a tangent and sweeps round the casing, the moisture by the whirling motion being thrown against the side, and the dry steam escaping down the central pipe.

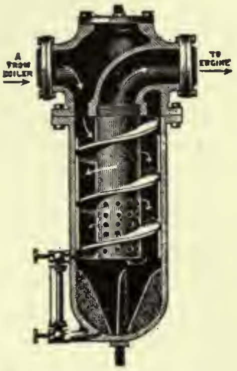


FIG. 191.—Centrifugal Separator.

Fig. 191 represents another separator on the same principle, a screw thread giving the necessary whirling motion to throw off the water.

No separator can entirely free saturated steam from water, they merely reduce it to a small percentage, which is carried along with the steam in exceedingly fine particles like mist or cloud. As the water collects in the separator it must be discharged automatically without loss of steam. For this purpose an apparatus is attached called a steam trap, which allows water to escape, but traps the steam.

Steam traps are of importance on the jackets of engine cylinders, on evaporators, in fact, on all steam apparatus where condensation takes place. Some work by means of a float lifted by the collected water, others by the differential expansion of tubes. When steam fills the tube it expands and closes the outlet valve; when the tube fills with water condensed, the water being colder than the steam, the tube contracts and the valve opens, discharging the water. Another one recently introduced works positively by a displacement piston or plunger.

In most cases traps discharge the water into a drain or tank, but in other cases the water can be returned to the boiler direct from the trap. The trap for this purpose is called a return steam trap.

A fourth type, acting with a spring diaphragm, both separates the water and traps the steam.

THE FLOAT STEAM TRAP

This type, made by Messrs. Richardson & Co., Hartlepool, is shown in Figs. 192, 193, 194 in section, and as applied to jacketed cylinders and steam chests on a marine engine in Fig. 195.

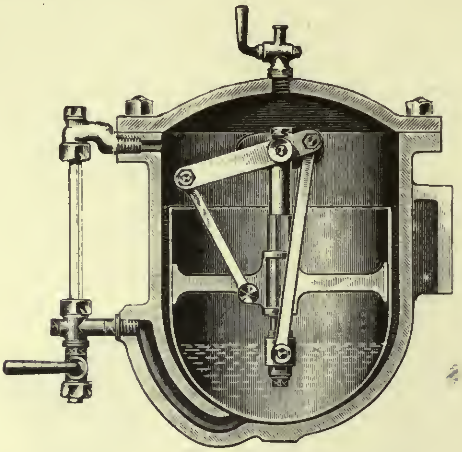


Fig. 192.—Float Separator or Trap.

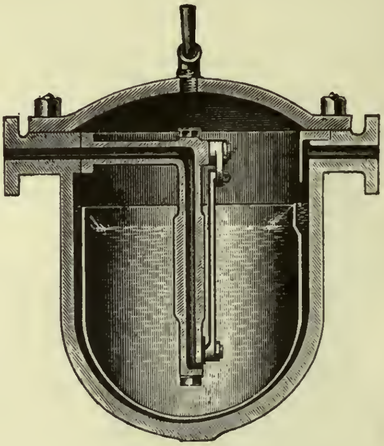


Fig. 193.—Float Trap.

The float is a metallic cup guided on a sleeve sliding upon a fixed water discharge tube down the centre. At the lower end of the discharge tube a slotted port is opened and closed by a sleeve valve operated by the rise and fall of the float, through two links and a lever. It will be gathered from the illustrations that as the float falls the valve opens, and as the float rises the valve closes. The water, as shown in Fig. 193, enters by the right-hand inlet, and is guided to fall between the casing and the cup.

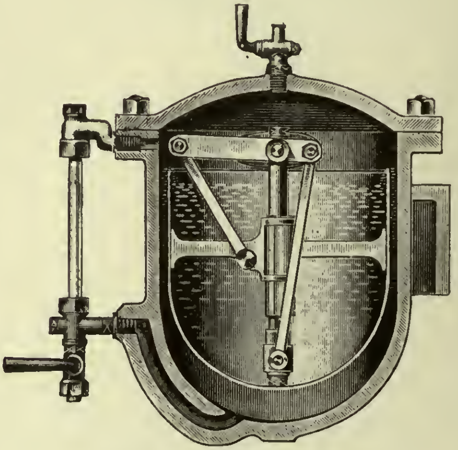


Fig. 194.—Float Trap.

The cup being, say, empty to start with, and at its highest position, the outside water floating it up, the discharge valve is closed; the water which comes in overflows from the outside of the cup to the inside as shown in Fig. 193, and sinks the cup until the sleeve valve opens the port at the bottom of the central tube, and the water is discharged from the cup until it rises again far enough to close the valve. The other two Figures show the cup at the upper position and lower position.

Fig. 196 shows an American form of float trap. The float is a hollow ball which by levers shown opens and shuts a balanced valve. It has a quick closing motion by means of the steam pressure. For this purpose a passage is made over the valve communicating with the outlet above the level of the valve passage. If this should fall slightly below the position shown in Fig. 196, steam will enter the upper passage and get behind the valve and help to close it.

A float valve for returning the water to the boiler is shown in Figs. 197, 198, known as Southwick's valve. Fig. 198 shows the valve with one-half of the casing removed; the section shows the mechanism not seen in the other Figure.

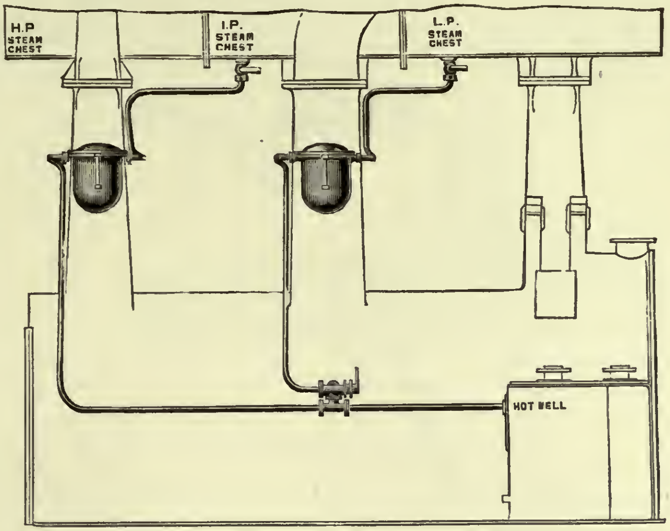


Fig. 195.—Trap applied to Marine Engine.

The buoyancy of the float is made to tilt a cradle holding an iron ball. The cradle is rigidly supported on a horizontal shaft at its centre, and on the same shaft is a lug which moves a side valve for admitting steam to the upper part of the trap. As the trap fills with condensation the float rises, bringing the cradle to a horizontal position. When such a position is reached that the cradle tips the ball rolls to the other end, acting as a positive gravity force to open the valve admitting steam. This drives out the condensation through the discharge pipe. The entrance and discharge pipes both have check valves, so that the motion of liquid can only be in one direction; and as the trap is mounted higher than the top of the boiler and uses steam at boiler pressure, the effective head to force the water into the boiler is that of the column of water in the trap and piping above the water level in the boiler. As the waters lower in the trap the float drops with it and tips the cradle and ball back to their former position, closing the side valve to live steam, and opening the vent ports through the same. The interior of the float is open to the atmosphere through the arm and shaft, as shown in the sectional drawing, and an accumulation of moisture inside of it is quite impossible. The sectional drawing (Fig. 197) shows correctly the principles of this type of trap.

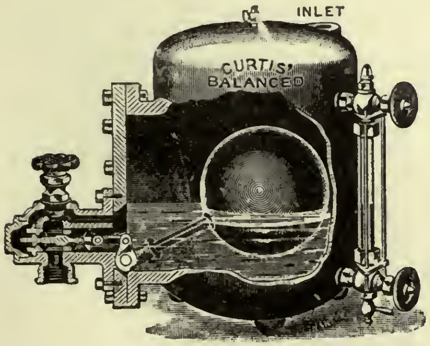


Fig. 196—Ball Float Trap.

THE STEAM LOOP

This is a device for returning the water separated from the steam in a separator to the boiler automatically. It depends for its action upon the syphon principle. In

Fig. 199, S is the steam separator, R the rising pipe, F the drop pipe ; the water is returned through the check valve C on the boiler. When water collects in the separator,

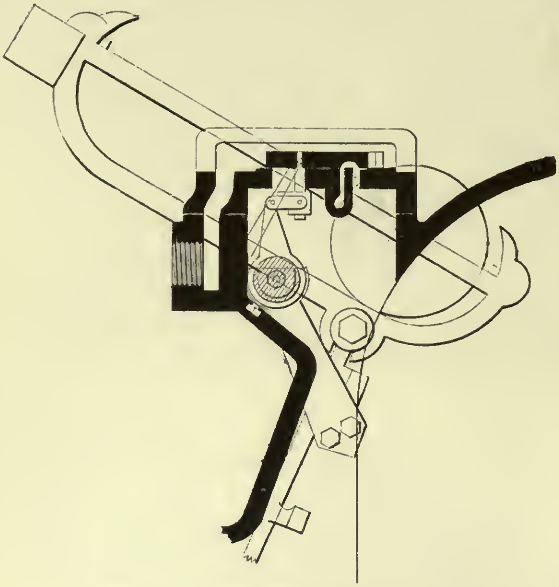


Fig. 197.—Return Steam Trap.

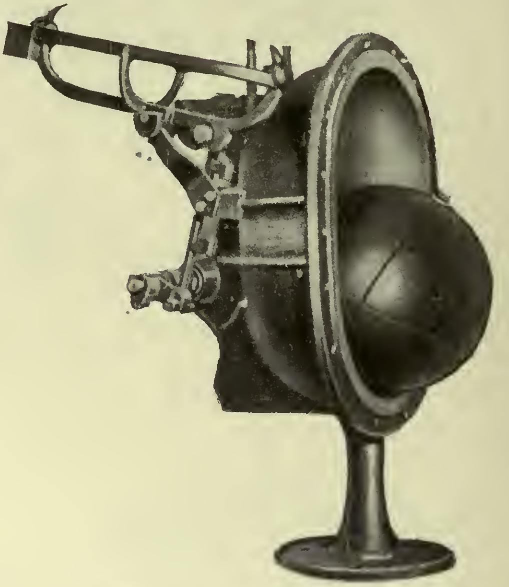


Fig. 198.—Return Steam Trap.

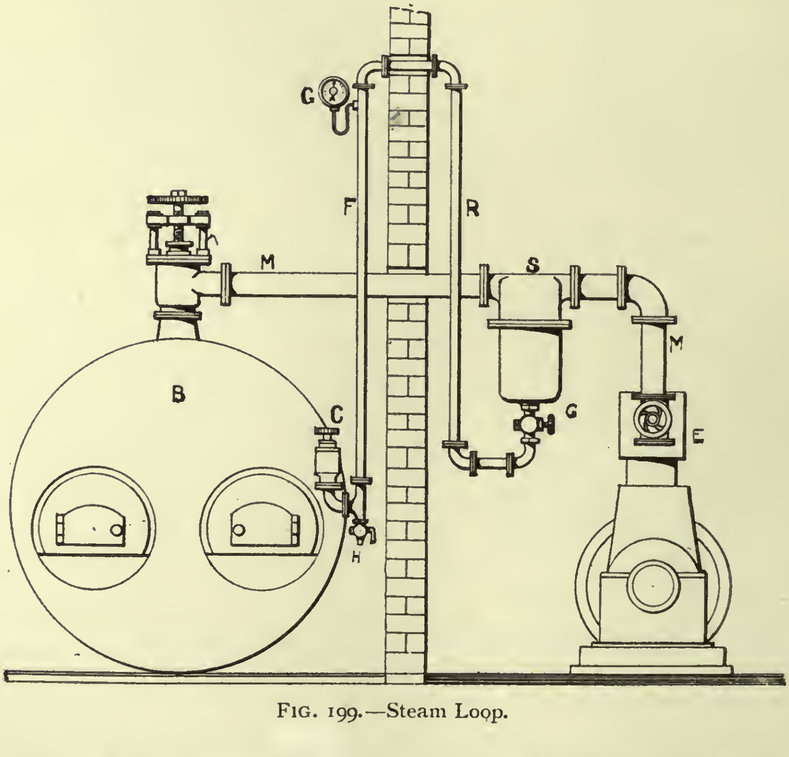


Fig. 199.—Steam Loop.

let the valves G and H be opened to blow out the pipes RF. Upon closing the valve H water will condense in F until the weight of the column opens the check valve C, the pressure in the boiler and separator being equal ; the weight of water in the drop leg opening the valve intermittently, and the water in the separator is blown up the riser pipe R, when the valve opens and the water rushes down F.

EXPANSION TRAPS

The type illustrated in Fig. 200 is one of the simplest, and has the advantage over most others in occupying less space, weighing less, and being more easily adjusted and repaired. The trap works on the principle of the unequal expansion of two different metal tubes. It consists of a bronze and an iron tube, which form two sides of a triangle, the frame of the trap forming the base. Inlet is by the bronze tube and outlet by the iron tube. So soon as steam enters the bronze tube it expands and closes the valve until water has formed and contracted it, when it opens again.

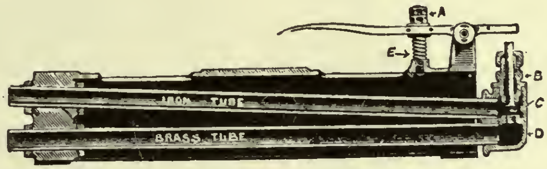


FIG. 200.—Expansion Steam Trap.

The action of the trap is very positive, and, when properly set, the passage of live steam is stopped. It has no internal levers, floats, or working parts boxed in and exposed to the action of the steam or water. It will work equally well on cylinder or steam pipe, and in any position. There are no faced joints to maintain steam-tight. The valve is always open to atmosphere, except when steam is passing; therefore water discharges as fast as it collects, even before steam is turned on. A clear blow through is obtained by pressing down lever. The valve is easily examined and replaced.

When connecting up the trap it is desirable to arrange for a stop valve on inlet and outlet pipes, also a bypass, so that the trap can be examined at any time while steam is on the pipes, without causing stoppage or inconvenience.

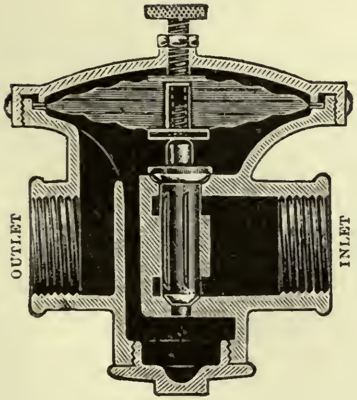


FIG. 201.—Diaphragm Steam Trap.

DUNHAM DRAIN AND TRAP

In this design, shown in Fig. 201, the pipe connections are now in one straight line, so that it can be placed in a line of piping without special couplings. It operates on a novel principle. The trap has an expansible metallic corrugated casing or receptacle, like that in aneroid barometers. This case is filled with a volatile liquid, which expands when heated by steam, and closes the water escape valve; but when water collects, it being colder than the steam, the case contracts and opens the valve. The edges of the case were soldered at first; they are now brazed. It can be used in vacuum pipes to remove condensation water by a special arrangement.

THE CENTRAL MARINE ENGINEERING TRAP

This firm, dissatisfied with the traps in common use, and finding great difficulty in regulating the condensed steam in the jackets of their cylinders, and in other parts of the engine where there is a chance of water accumulating, and the application of several of the advertised drain traps not being sufficient to get over this difficulty, they were consequently forced to think out a trap for themselves, and the one illustrated in Fig. 202 is one which, as may be seen, is a positive-action trap, and it has in every application of it given entire satisfaction. In fact, if it is kept in order by the attendants it cannot possibly fail to operate. It consists of an outer chamber, in which is a piston operated by some reciprocating part of the engine. When the outer chamber becomes full of water from the drains this water percolates through the perforations in the movable sleeve; the rising piston imprisons this water, forces up the sleeve, which, being attached to the bottom valve, releases the water. So long as water remains in the outer chamber this operation will be repeated, but the moment the chamber has been drained and steam only is operating the valve remains closed.

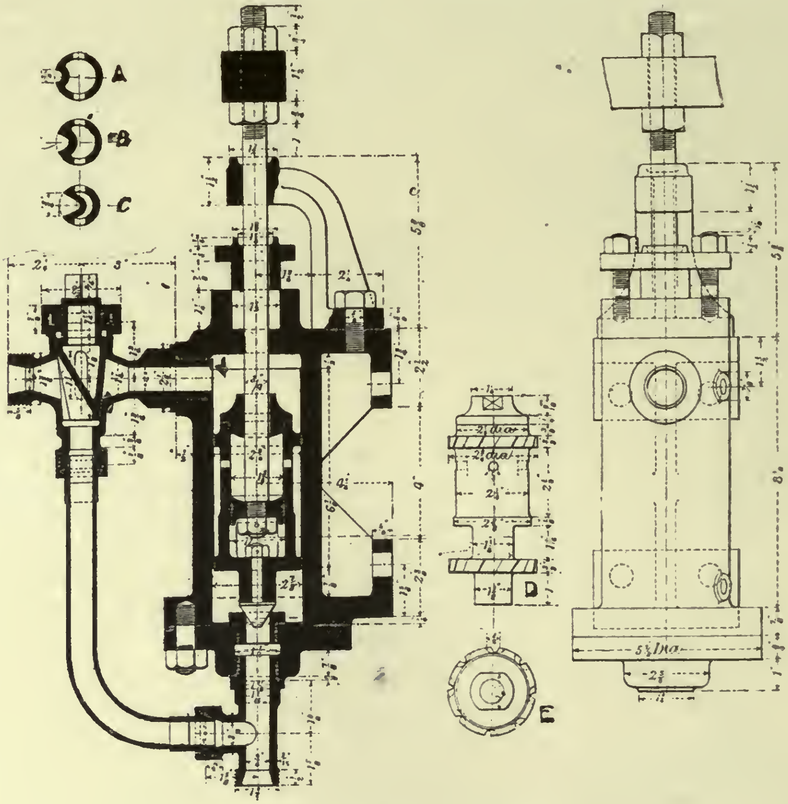


Fig. 202.—Positive Steam Trap.

The details of this interesting trap are shown fully in Fig. 202.

MORISON'S MARINE EVAPORATORS

Mr. Morison, the inventor, has pointed out the advantage of this apparatus in his paper on the subject. They are here again set forth.

The universal practice on steamships a few years ago was to supply the loss of water, due to leakage in a condensing engine, from the sea direct to the boilers; but since the introduction of multiple expansion, with its higher pressures and temperatures, the necessity for minimising the scale in boilers has resulted in the use of an independent apparatus, in which sea water is evaporated by means of steam, and the generated steam, being condensed, is delivered to the boilers free from all impurities. Apart from the necessity of such an arrangement from an engineering point of view, there is the commercial consideration of reducing the boiler scaling expenses, minimising the liability to accidents, and prolonging the life of the boiler; as whilst it is possible for careful engineers of long experience to work boilers without an evaporator, there are few superintendents who do not realise the fact, that in order to obtain the greatest engineering and commercial efficiency, and as a safeguard against accidents, an evaporator is a necessary auxiliary on board a modern steamship.

Steam traps, returning the condensed water from all places where it may collect, do much to reduce leakage of water.

The losses of water in marine engines is often unnecessarily extravagant. A case is quoted of the engines being in such a bad condition that no less than 20 tons of water was lost in 24 hours by a plant of 1000 horse-power indicated. Whereas in another case 2½ tons per 1000 horse-power was lost in 24 hours. These are two extreme cases as to the quantity of auxiliary feed water.

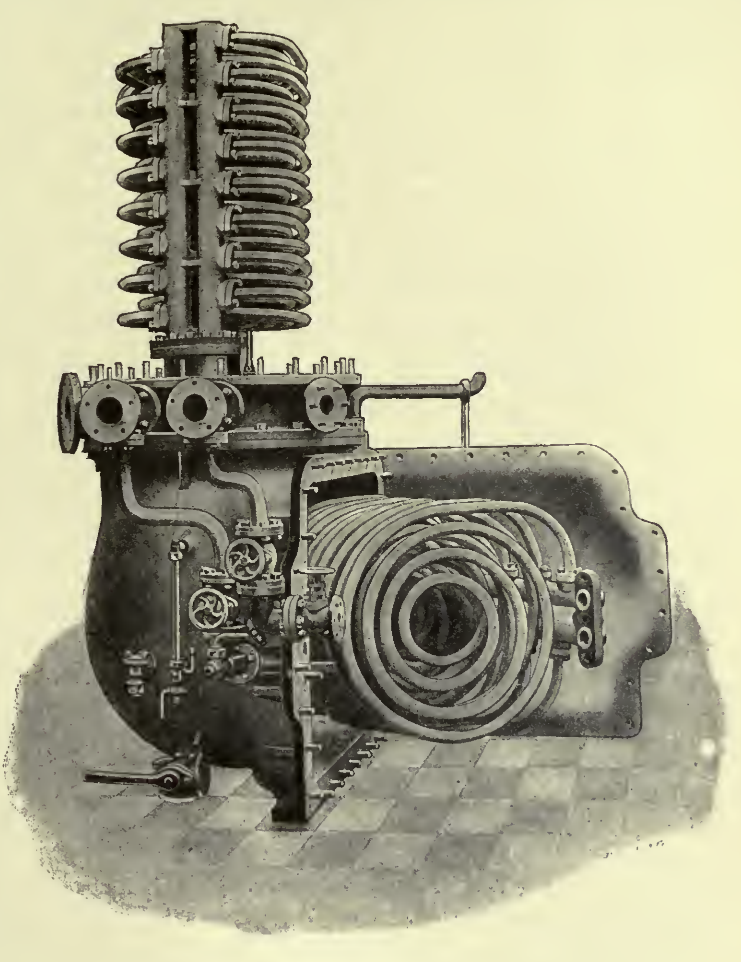


FIG. 203.—Morison's Evaporator and Condenser.

The Committee on Naval Boilers recommend in their report that not less than 6 tons per 1000 indicated horse-power be allowed. This allowance seems very small when the large number of auxiliaries is considered, but probably the estimate is based on the maximum indicated horse-power, which is, of course, seldom required, and it must also be remembered that the machinery in His Majesty's Navy is kept in the highest possible condition of efficiency. It will be evident, however, that not only does the amount of auxiliary feed vary for different types of engines, but the variation may be very great in ships of the same class, depending entirely on the condition of the main and auxiliary machinery. In passenger boats, where the labour is sufficient to keep everything in good order, a safe allowance is 8 tons per 1000 indicated horse-power at sea, which, assuming a consumption of 15 lbs. of water per indicated horse-power per hour, is 5 per cent. of the total feed; but in cargo boats, where the available labour is much, the probable cost of obtaining fresh water by means of an evaporator is a most important consideration. There are several arrangements available: as steam, being taken from the boiler direct, the steam generated in the evaporator may be led to the steam chest of the low-pressure cylinder, to the main condenser, or to the hot well. In order to compare the relative economy of these methods, loss by radiation may be neglected, as it is practically a constant quantity; also for the sake of simplicity the usual tables on the properties of steam may be taken as applicable to the general conditions.

The cost is calculated as follows when the steam is supplied to the evaporator from boiler, and steam generated in evaporator discharged to hot well.

In considering the method by which steam is supplied to the evaporator direct from the boiler, and the steam generated in the evaporator is discharged to the hot well, or its equivalent, and there condensed amongst the feed water, let it be assumed that the steam pressure in the evaporator is 1 lb. per square inch above the atmosphere, or say 16 lbs. absolute pressure, that the water fed to the evaporator is drawn from the circulating pump discharge at a temperature of 80° Fahr., and that the temperature of the hot well is 120° Fahr.

Experience has shown that marine evaporators should be worked at a density of from $\frac{3}{32}$ to $\frac{4}{32}$, but for purposes of calculation assume $\frac{3}{32}$; then, in order to maintain that density in the evaporator, one-third of the total amount of sea water fed to the evaporator must be discharged into the bilges or overboard. Consequently, to produce 1 lb. of pure steam $1\frac{1}{2}$ lb. of sea water must be supplied to the evaporator, 1 lb. of which is evaporated into steam, and the remaining $\frac{1}{2}$ lb. is discharged as hot brine. Therefore the total heat required to make 1 lb. of pure steam is the sum of the heat in the steam and that in the discharged brine.

Temperature of steam at 66 lbs. absolute	. . .	216°.3 Fahr.
Heat in 1 lb. of steam at 216°.3 above 80°	. . .	1099.4 ⊖
Heat in $\frac{1}{2}$ lb. of brine at 216°.3 above 80°	. . .	68.1 ⊖
Total heat required to produce 1 lb. of steam	=	1167.5 ⊖

If the boiler pressure is 175 lbs. absolute and temperature 370°.8 Fahr., and it leaves the evaporator at 250°.4 Fahr., then the heat given up by each pound of boiler steam will be 976 ⊖; and as 1167.5 ⊖ are required to produce 1 lb. of water pure in the evaporator, this is equivalent to $\frac{1167.5}{976} = 1.19$ lbs. of steam taken from the boiler to produce 1 lb. of fresh water.

But as the pound of steam generated carries its heat and condenses into the hot well it gives up all its heat above the hot well temperature $= 1099.8 - (120 - 80) = 1059.4$ ⊖. The drain from the evaporator coils is also delivered to the hot well, giving up $(250.4 - 120) \times 1.19 = 156$ ⊖, so that the net cost of producing 1 lb. of fresh water is that due to "brining" $= 68.1$ ⊖.

The quantity of coal required may be found by taking 1 lb. of good coal at a value of 14500 ⊖, boiler efficiency 66.6 per cent. We get $\frac{14500 \times .666}{966.6} = 10$ lbs. of water evaporated from and at 212° Fahr. per pound of coal, and as 9666 ⊖ are available per pound of coal burned, the pure water produced by 1 lb. of coal will be $\frac{9666}{68.1} = 141.9$ lbs., so that $\frac{2260}{141.9} = 15.7$ lbs. of coal per ton of pure water produced.

In the other method, steam supplied to the evaporator from a boiler, and the steam generated in the evaporator is discharged to the receiver of the low-pressure engine.

In this case the generated steam from the evaporator is discharged into the steam chest of the low-pressure cylinder, and does work upon the low-pressure piston before it is finally condensed.

Again referring to the data contained in the report of the committee already mentioned, it is seen that in economical triple-expansion engines about 17 per cent. of the total heat in the steam is converted into work. For the sake of simplicity, however, we will consider the efficiency as 18 per cent., and, assuming that the power developed in each cylinder is equal, one-third of this, *i.e.* 6 per cent. of the total heat, is utilised in each cylinder. It therefore follows that $(100-18)=82$ per cent. of the total heat is rejected; further, that the amount entering the steam chest of the low-pressure cylinder will be 88 per cent. of the total supplied to the engine, and of the total heat supplied to the low-pressure cylinder $\frac{6 \times 100}{88}=6.82$ per cent. is converted into work.

Let the pressure in the steam chest of the low-pressure cylinder be 7 lbs. per square inch, or say 22 lbs. absolute pressure, and the temperature of the evaporator feed 80° Fahr., as in the previous case.

Temperature of steam at 22 lbs. absolute	233°.1 Fahr.
Heat in 1 lb. of steam at a temperature of 233°.1 Fahr. from water at 80° Fahr.	1104.5 ⊖
Heat in $\frac{1}{2}$ lb. of brine at a temperature of 233°.1 Fahr. from water at 80° Fahr.	76.5 ⊖

Heat required to produce 1 lb. of pure steam $=1104.5+76.5=1181.0$ ⊖

The equivalent weight of boiler steam necessary to produce the pound of evaporator steam may be found in a similar manner to the first, with this difference, that as the working pressure in the evaporator is higher, the drain from the coils must also be at a higher temperature; this in practice is generally about 292°.7 Fahr., or the temperature corresponding to a pressure of 60 lbs. absolute; then the heat given up per pound of boiler steam at 160 lbs. pressure and a temperature of 370°.8 Fahr. above 292°.7 Fahr. = 933.7 ⊖, so that $\frac{1181}{933.7}=1.26$ lbs. of boiler steam is necessary to produce 1 lb. of evaporator steam. And taking 120° Fahr. hot well temperature as in the previous case, the available heat per pound of boiler steam = 1106.4 ⊖, or $1406.6 \times 1.26=1398.5$ ⊖ are passed to the evaporator instead of being employed in the engine in the performance of work, in which case 18 per cent. would have been converted into work as already explained, this amounting to $1398.5 \times .18=251.7$ ⊖.

We have also seen that 6.82 per cent. of the heat received by the low-pressure engine is capable of conversion into work therein. It therefore follows that 6.82 per cent. of the heat (above 120° Fahr.) contained in the steam discharged from the evaporator to the low-pressure casing, or $(1104.5-40) \times 0.0682=72.6$ ⊖ are convertible into work, as compared with 251.7 ⊖ if the boiler steam had been passed through the engine direct instead of to the evaporator, and thence to low-pressure engine. Then of the total useful heat $\frac{72.6}{251.7} \times 100=28.8$ per cent. can be utilised in the low-pressure engine. The evaporator has given up 28.8 per cent. of the total available heat above 120° Fahr. in the steam generated therein $=(1104.5-40) \times 0.288=306.8$ ⊖, the remaining 757.5 ⊖ being lost along with the ⊖ lost in brining. The heat in the water drained from the coils goes, however, to the hot well, and amounts to $(292.7-120) \times 1.26=217.5$ ⊖.

Fig. 203 shows the larger evaporator to be employed in cases wherein evaporators are sometimes used as condensers for winches when in port, and certainly seem well adapted for the purpose. The cooling surface required is usually in excess of what is necessary for evaporating purposes, consequently an increased size of evaporator has to be fitted.

An objection to a combined evaporator and winch condenser is that it necessarily contains very much more surface than is required for an evaporator only, and it is therefore much larger and more costly. But there is the practical objection that on arrival in port an evaporator should be cleaned, yet it is on arrival in port that it is required as a condenser, and the natural result follows that it is often set to work as a condenser with scale on the coils, and therefore its efficiency as a condenser is greatly reduced.

An entirely independent condenser, however, is always in a condition to condense all the steam, and, requiring no attention whatever, it can be placed where most convenient for a simple lead of pipes. A most important advantage in a separate condenser is that it gives the engineers time and opportunity for cleaning the evaporator, and each apparatus is therefore maintained in a state of maximum efficiency.

Fig. 203 shows a combined apparatus which consists essentially of a radial evaporator with a division chamber between the base and the dome, on which the cooling coils are mounted. As will be seen from the illustrations, the coils in the upper and lower portions are similarly attached to inlet and outlet tubes, the former being arranged horizontally. The amount of surface in the lower or evaporator coils is arranged to produce the desired quantity of auxiliary feed water, and, when evaporating, steam is admitted to the lower coils only. When condensing, both the upper and lower coils are used, the connections being arranged to admit the circulating water inside the tubes, the steam from the winches, etc., being led directly into the dome. The dome is provided with a cover and also a door on the side, by means of which the various joints and connections may be tested and examined. The design is specially applicable to cases where the head room is limited, as the coils may be disconnected from the standards by means of the door on the side and removed through the upper cover, or the connections between the inlet and outlet tubes or standards may be broken and the whole of the upper portion lowered into any convenient position for disengagement.

STEAM PRESSURE REGULATING OR REDUCING VALVES

It is often necessary to work engines at a lower pressure than that of the boiler, and for this purpose automatic valves are employed whereby the pressure can be maintained at a constant pressure below that of the boiler.

And a further use for them is to maintain a constant pressure under varying boiler pressure; but, of course, the variations of the boiler pressure must be all above the constant working pressure required. Suppose 150 lbs. pressure is the constant pressure required, then the boiler may be set for a maximum pressure of 180 lbs., the reducing valve for 150 lbs., in which case the stokers may vary the boiler pressure from 150 to 180 lbs. per square inch without affecting the engines.

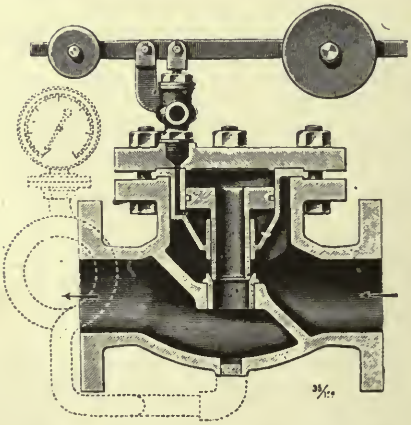


FIG. 204.—Dead Weight Reducing Valve.

The reducing valve ought to be as close as possible to the steam engine inlet, so as to secure as much as possible the benefit of any superheating due to the "wire-drawing" of the steam by the reducing valve. Its proper place is between the water separator and the engine.

This superheating of steam by wire-drawing has been practised in some cases to supply dry steam instead of the tubular superheaters in the flues. The practice has been adopted for some years, apparently with success, in the Royal Navy. For instance, the engines of the *Terrible* were designed to work at a pressure of 210 lbs. at the stop valve, but steam is generated in the boilers at 250 lbs. In many cases in the Navy the steam is generated in the boilers at 300 lbs. per square inch, and is used at 210 lbs. at the engine stop valve.

Again, in some cases the pressure to which an engine can be worked is lower than that to which the boiler can be worked; not a common case, but sometimes it does occur. In such a case it has been found that at a reduced boiler pressure to suit the engine the boiler could not keep steam enough; but by raising the boiler pressure by 30 lbs. and reducing the pressure to 70 by a reducing valve, the boiler could easily do the work. In the first case the steam from the boiler carried over water, and more formed by condensing in the pipes. In the second case the higher-pressure steam carries less priming water, and condenses less; and at the reducing valve the "wire-drawing" superheats and dries the steam.

There are many reducing valves in the market. The Körting valve is shown in Fig. 204 as an example. It is a dead-weight type.

The valve consists of a tube, free to move in a sleeve and provided with a piston working in a cylinder, the bottom of the tube constituting the upper valve face. The volume of steam which passes is fixed by the valve lift, and is varied by the up-and-down movement of the piston.

The reduced steam pressure is on the top of the piston, and high-pressure steam which leaks past the sleeve is on the bottom. The latter pressure is constant, as any excess pressure is relieved by a small escape valve connected with the space under the piston. The pressure can be varied by altering the position of the lever weight on the relief valve. The piston therefore rises or falls with variations in pressure of the reduced steam. If, for any reason, this is less than the relief valve pressure the piston rises, allowing more steam to pass the valve, and restoring the balance of pressure between the upper and lower side of the piston. Similarly, the piston is forced downwards with excess of pressure, and the volume of steam passing the valve is reduced.

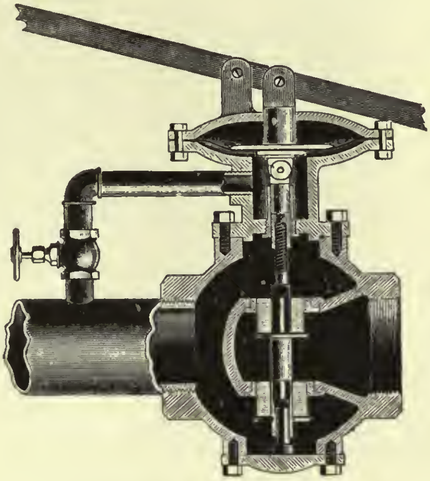


Fig. 205.—Körting Reducing Valve.

The valve possesses the following important features:—

It reduces steam from any pressure to any desired lower pressure, and the working does not depend in any way on the quantity of steam passing, or the initial pressure of this.

The pressure of the reduced steam can be altered during working.

As the valve is simple in construction and has few parts it is not liable to get out of order.

The direction of flow of the steam is indicated by the arrows on the illustration (Fig. 204).

In another form this reducing valve is designed with a diaphragm, and a balanced valve, as shown in Fig. 205. The lever is weighted to adjust the pressure.

For marine purposes the valves are spring loaded, instead of by weights and levers.

MORISON'S EVAPORATOR FEED HEATER

This feed water heater has been designed with the view of overcoming the objections to evaporating directly into the hot well. The feed heater is shown at Fig. 206, and consists simply of a cast-iron vessel into which the steam from the evaporator is admitted, and, becoming condensed amongst the feed water, imparts its heat to the latter. The water from the hot well enters the heater by CW, and the steam from the evaporator by SM. The slots in the internal pipe SM being set at an angle, the escape of the steam causes a centrifugal motion of the water, among which it is rapidly condensed. The heated water flows out at HW to the feed pump suction. Any steam or vapour generated within the heater or any air freed from the water is allowed to escape by AO to the hot well, the connection being made at any point above the water level. The feed heater should be placed so as to divide the distance from the air pump to the feed pump suction into two equal parts, so that the vertical distance from the air pump head valve to the feed heater will be about equal to the vertical distance from the feed heater to the feed pump suction. In some cases, however, where the hot well is of small capacity, it will be found of advantage to place the feed heater at a lower position than that described. The condensing of the steam among the feed water naturally raises the temperature of the latter, but in ordinary cases not to such a point as will influence the working of the feed pumps, unless the feed pump valves are exceptionally heavy, or the design very faulty. If any difficulty occurs a remedy is often found by making a connection by a $\frac{3}{8}$-inch pipe between the top of the feed pump barrel and the condenser. This system of feed heating naturally has a limit, although for ordinary cases, and especially in cargo boats, it can be successfully applied.

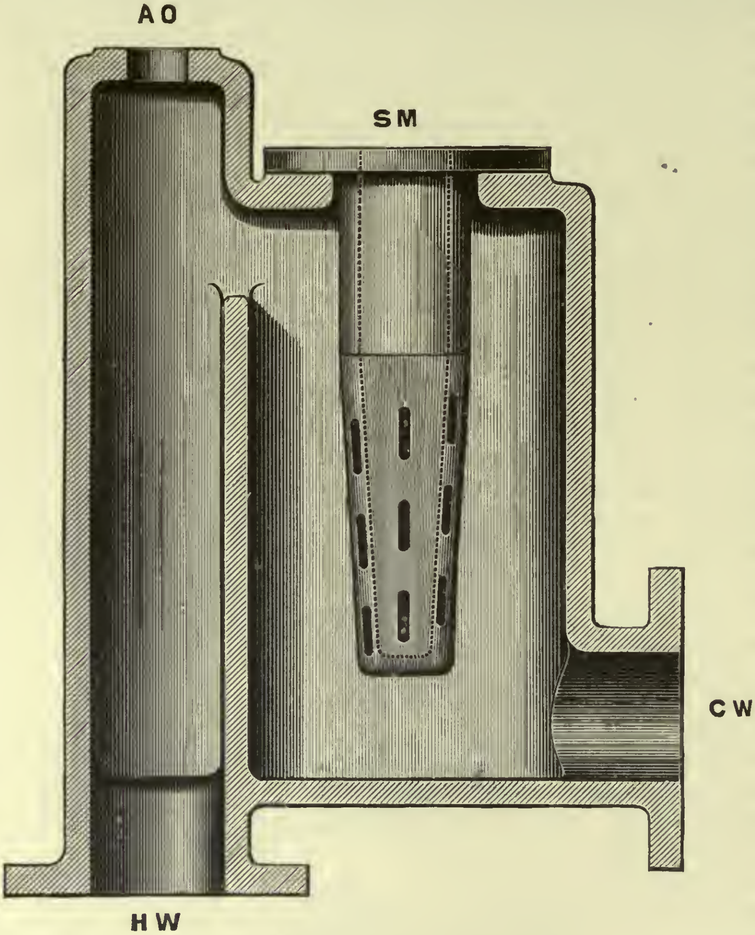


Fig. 206.—Morison's Feed Water Heater.

ARRANGEMENTS OF FITTING EVAPORATORS

Evaporating into the Main Condenser and Evaporator Feed Heater.—Fig. 207 illustrates the arrangement recommended for fitting an evaporator on board ship. For clearness of description the position of some of the mountings has been slightly altered in the drawings from the actual position on the evaporator.

Steam direct from the main boilers enters the heating coils through the valve J, and the resulting water of condensation is drained away by the pipe X, which leads into pipe C_1 at any point beyond the 2-way cock as shown. The sea water is taken from the circulating discharge by the pipe R, and supplied to the evaporator by means of the small pump V, through the pipe W, and non-return valve G. The generated steam leaves the evaporator by pipe S, and may be discharged by means of the 2-way cock Y, either into the main condenser through the reducing valve T, or into the feed heater FH through pipe C_1. The water from the hot well enters the feed heater by the pipe A_1, and serves to condense the steam generated in the evaporator, and after being raised in temperature flows by pipe B_1 to the suction valves of the main feed pumps. The air and vapour from the feed water escape from the feed heater by pipe D_1 into the hot well air vessel, the pipe being connected at any point above the water level in the hot well. The brine from the evaporator is discharged to the bilge through the brining cock M and pipe U.

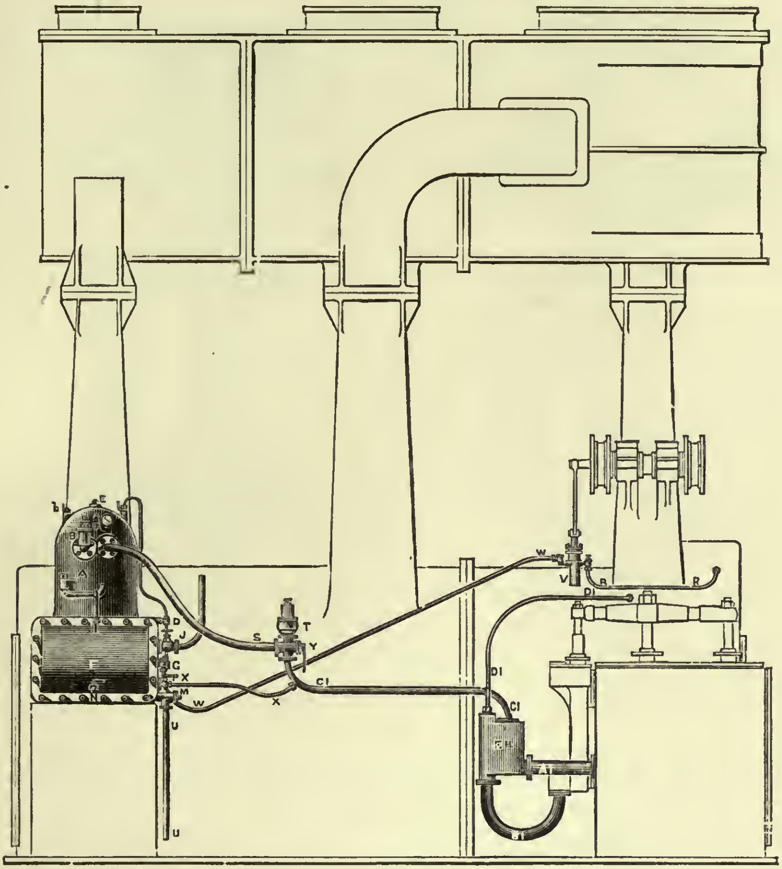


FIG. 207.—Fitting Evaporators to Marine Engine.

AULD'S REDUCING VALVE

A spring loaded reducing valve suitable for marine and other boilers is shown in Fig. 208. A is the inlet and B the outlet. A set screw abuts against the top of the spring, whereby the pressure can be accurately adjusted to a scale and pointer on the upper end of the spring, as clearly shown.

Reducing valves are made either fitted with springs, or with levers and weights, as may be considered most suitable for the purposes to which they are applied, and having a graduated scale, can be set easily to the reduced pressure wanted. They are also applicable for reducing and regulating the pressure of water or air.

Where reducing valves are used for supplying steam to weak vessels without safety valves, a safety valve should be placed on the low-pressure side of the reducer.

In cases where reducing valves are used for supplying steam from a high to a low-pressure boiler it is advisable to have an extra safety valve on the low-pressure side in addition to the boiler safety valve.

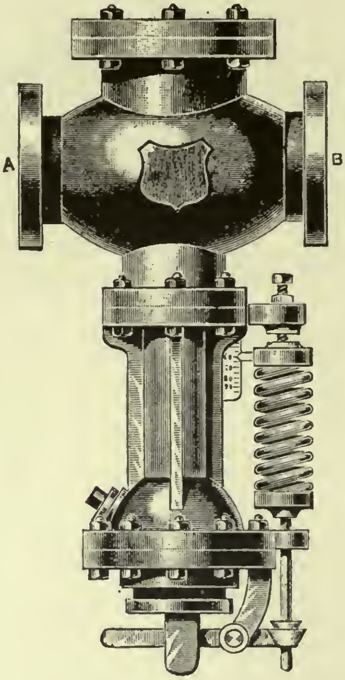


FIG. 208.—External View.

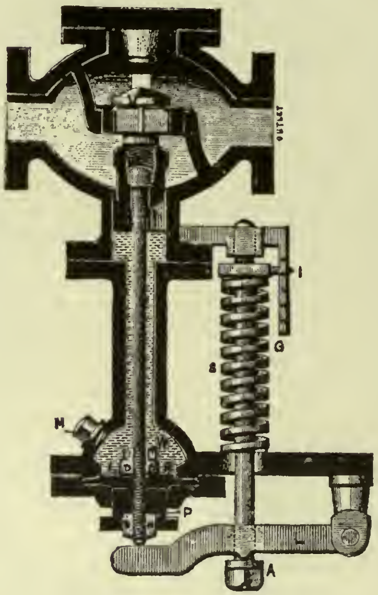


FIG. 209.—Section.

Auld's Reducing Valve.

It very frequently occurs that a plant of boilers is put down to supply high-pressure steam to engines, and that steam is also required at a lower pressure for boiling or heating purposes. In such cases a reducing valve is desirable, and the type here described is a very suitable one.

SUPERHEATERS

The competition of the internal combustion engine on land, and the keen growing competition for over-sea freights by steamship owners, has compelled attention to economising steam in steam engines in every way possible. Compounding and condensing go far in economical working. The first, by reducing the range of temperature in the cylinders; and the latter, by lowering the temperature at which the exhaust is rejected, the heat engine efficiency being proportional to two temperatures T, T_1,—the first being the temperature of the fluid entering the engine, the second the temperature at which it leaves the engine. The energy available theoretically is proportional to $\frac{T - T_1}{T}$, we can increase the energy available by lowering T_1 or increasing T.

The chief effect of superheating is to provide dry steam right up to the point of cut-off in the cylinder of the engine, or in the wheels of the steam turbine.

Superheating is of more advantage in the piston engine than in the turbine. In the former there are larger surfaces for cooling the steam; while in the turbine not only are surfaces less, but the energy of the steam to a large extent depends on its weight per cubic foot multiplied by its velocity. Superheating increases the volume of the steam at constant pressure; hence its weight is reduced, and consequently its reaction on the turbine blades is reduced. Moderate superheating of steam for turbines is beneficial only in providing dry steam. If water is mixed with the steam in a turbine its harmful effects are chiefly due to its friction in the narrow tortuous passages between the turbine blades.

A hot air jacket on a steam turbine might easily be provided to keep the steam dry enough to the end of its expansion without any extra superheating.

In engines designed for a high rate of revolutions the cylinders are small compared with those of slowly revolving engines, so that at first sight one would conclude that superheating would prove more beneficial with the latter type, but experimental evidence so far indicates that the quick revolving engines profit most by it.

The saving in steam effected by superheating is undoubtedly very considerable. 175° superheat in a good engine has shown a drop in steam consumption from 25 lbs. per horse-power-hour to 15 lbs. with the superheat. This saving is, however, not all gain. Superheating requires heat, which means fuel consumption, and this raises the question of greatest importance as to the design of superheaters in which the superheating is effected with the smallest consumption of coal or fuel.

Marvellous records of cases are given where the adoption of superheating has enabled nearly twice the work to be done by the same boiler power. They may very well be true, but the question is, How much fuel did they save compared with what would have been consumed if the boiler power had been doubled, instead of adopting superheating? However, there is no doubt there would be a saving in fuel with superheaters in nearly all cases.

Experience alone will finally decide the best construction of superheaters. Some of them have been placed in the boiler flues, others have independent firing. The boiler flue type has some drawbacks in the way of regulating the temperatures, but otherwise it is a cheap system, utilising the waste heat of the furnaces. It is held, however, by many engineers that it is better to absorb all the available heat in the boilers and feed water heating, and to employ a well regulated fire for superheating separately. On this point it may be observed that the design of fire or furnace in a separately fired superheater is of considerable importance. For heating by combustion we may divide furnaces into three classes. First, the "boiler furnace." In this design the fuel and gases all burn within the furnace, and the heat given to the boiler and its tubes is delivered in two methods: first, radiant heat from the incandescent and hot fuel on the fire-grate strikes the heating surfaces, and is absorbed nearly completely; secondly, the flame from the burning gases and the hot products of combustion impart their heat by radiation and contact effects; no system of boiler heating has given equally good results as this complete combustion of the fuel in the boiler furnace. For comparison, an illustration of an early Babcock & Wilcox boiler is shown in Fig. 210. This style of furnace has been tried on superheaters, but without much encouragement, for, unlike boilers, superheaters are not filled with a great mass of heat-absorbing fluid. The water in the boiler prevents any great heating of the plates and tubes, so that the fire may be strongly increased without overheating. The superheater, however, may be overheated, and the dry steam has little effect in carrying off any surplus of heat rapidly. It is therefore necessary to introduce some means of regulating or modifying the heat under variable conditions of the fire.

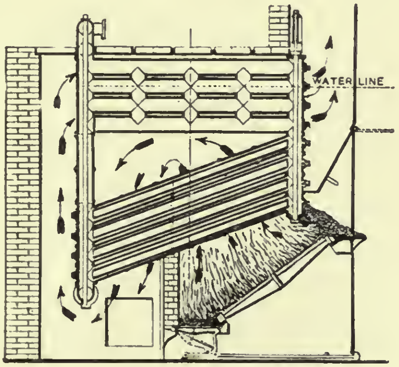


FIG. 210.—Boiler Furnace.

In Rosenthal's patent one method is proposed for this object—using a boiler furnace. It is shown in Fig. 211 combined with a boiler. It specially relates to superheaters and steam generators for marine use. The combined superheater and steam generator comprises a skeleton boiler of the Babcock & Wilcox marine type, consisting of steam drum A, downcomer tubes B terminating in a cross box C, inclined water-circulating tubes D connecting the cross box C to a cross box D^1, into which are expanded uprising tubes E connected to a cross box F, from which horizontal tubes G discharge the uprising water and steam into the drum A. The sides of the skeleton boiler are formed by inclined water tubes extending between end headers E, which are connected to the cross boxes D^1 and F at rear and to the cross box C and drum A in front. The superheater is composed of manifolds K located at one end of the boiler, the manifolds being connected alternately to inlet pipes L, into which is led, from the various boilers supplying it, the steam requiring to be superheated, and to outlet pipes L^1. The manifolds have expanded into their inner wall a series of U-shaped superheating tubes M. The superheater and the boiler elements are supported over a furnace chamber N. The furnace gases pass up between the rows of water tubes D, circulate around the superheater tubes M, and escape upwards between the steam and water tubes G to the uptake R. The fiercest heat of the furnace gases is reduced by their impingement on the rows of tubes which absorb part of the heat, and the superheater tubes M are thus subjected to the action of the gases at a lowered temperature, which increases their durability. The surrounding of the superheater by walls composed of water tubes permits of further absorption of heat for steam-generating purposes, and enables a suitable iron casing to be used, which is especially advantageous in superheaters fitted on board ship, and under conditions where it is impracticable to use brickwork walls.

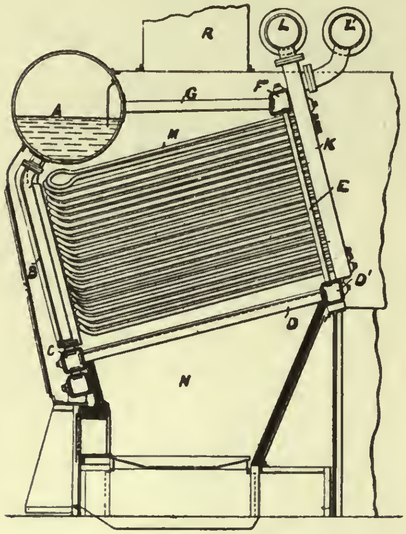


FIG. 211.—Boiler and Superheater.

Any superheater exposed to the radiations and flame and hot products of combustion combined must be very carefully fired to maintain the safe limit of heat rate supply. The flame and hot products if supplied alone can be regulated, but the radiant heat of the fire cannot be regulated. This leads to a design of furnace of the second type, the reverberatory furnace. Metallurgists and chemists at a very early date found the boiler type of furnace heating uncontrollable, and designed the reverberatory furnace, in which the fuel is burned in a fire apart from the object to be heated, the heat being carried over by flame and hot gases, which could be controlled to give the desired temperatures. The reverberatory furnace is shown in Fig. 212. This furnace has been adopted for superheaters, of which two types are here given.

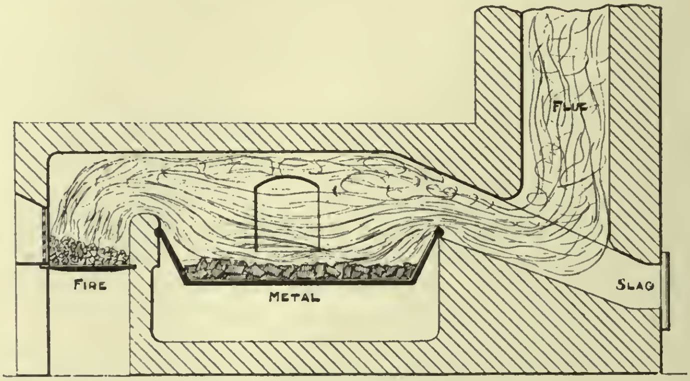


FIG. 212.—Reverberatory Furnace.

The first, an American type, shown in Fig. 213, as made by the American Steam Superheater Company. This superheater consists, as shown in the engraving, of a furnace formed entirely of brickwork, so that the combustion takes place away from any chilling influences, and the metallic surfaces are removed from the direct radiant heat of the fire and the first impingement of the gases. In the hollow bridge wall is a passage controlled by a register, through which air may be admitted to mingle with the gases as they pass from the combustion chamber to the superheater proper, thus tempering them to the degree of heat required.

The uniform moderate temperature of the whole body of heat-conveying gases, before their passage through the tube system, prevents sudden contraction and expansion, and effectually ensures against injury to the tubes, while the ease of regulation of the volume of the gases enables the degree of superheat to be held fairly constant at all times. The value of these features, ensuring as they do a constant high efficiency of the steam used in the engine, and a long life of the superheater tubes, and therefore a safe and durable superheater, cannot be too strongly emphasised.

The superheater proper consists of horizontal rows of straight tubes of standard commercial sizes expanded into steel headers, the joint and the front of the header being protected from the hot gases by non-conducting material. The openings in the headers opposite each tube are closed by removable plates held in place by bolt, crab, and clamp, the headers being spot-faced on the inside. The dead air space between the headers and the outer casings combined with the above-described insulation produces a degree of header temperature in operation but little in excess of that of the steam itself. Steam is delivered to the topmost row of headers on the left of the front elevation, and taken from the bottom of this row to the top of the first header of the opposite row, from the bottom of which it passes to the top of the second left-hand header, and so on, crossing the furnace repeatedly until it arrives at the bottom header on the left-hand side, whence it is drawn off for use. The hot gases entering the superheater at the bottom are deflected by rows of refractory tiling arranged as shown in the longitudinal elevation, so that it makes five passes lengthwise of the superheater before being discharged to the flue. The currents of steam and gas are thus opposed to each other, the coolest gas meeting the coolest incoming steam. By manipulation of the damper in the uptake and the register in the bridge wall it has been found that the desired degree of superheat can be closely maintained under conditions involving rapid fluctuations in the quantities of steam superheated. Drip connections from the lowest point in each header allow the water condensed while steam is being formed in the boiler to be drawn off. The sides

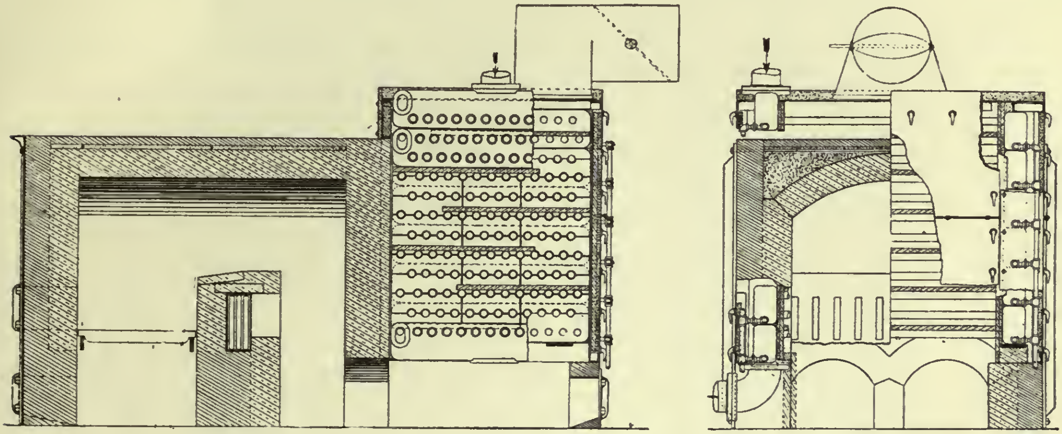


FIG. 213.—American Superheater.

and ends of the superheating chamber are formed of a removable casing, permitting access to the tube system.

Belonging to this type the Davey-Paxman superheater may be taken as a good example. It is shown in Fig. 214, where the furnace may be seen screened from the superheater by brick building. Thorough combustion takes place in the red-hot flues, so that the flame does not reach the tubes, but highly heated products of combustion at a fairly uniform temperature impinge upon the superheater. Figs. 215, 216 show side elevations of this superheater—the one for highly superheating a moderate amount of steam, the other for slightly superheating a large quantity of steam.

The header boxes are of cast malleable iron, and are divided into two compartments. The loop tubes are of solid drawn mild steel, have greater tensile strength at 600° Fahr. than when cold.

The course of the steam is indicated in the high temperature superheater by arrows (Fig. 216), and is first admitted at the lower header marked A; the arrows show the direction of the flow through the loop tubes B, in which the moisture in the saturated steam is wholly or partly evaporated. The steam leaves the upper part of the lowest header through an aperture connecting it up to the up-carrying tube C, which delivers the steam into the top header D, from whence it passes through the loop tubes E back again to the lower part of the header D, which is connected to the upper part of the header F by the double elbow G, and passes through the loop tubes H. This process of passing and repassing the steam through the loop tubes, headers, and elbows is continued until the steam reaches the outlet V, which it leaves in a more or less highly superheated state, according to the temperature of the superheating chamber and the quality and volume of steam passed through.

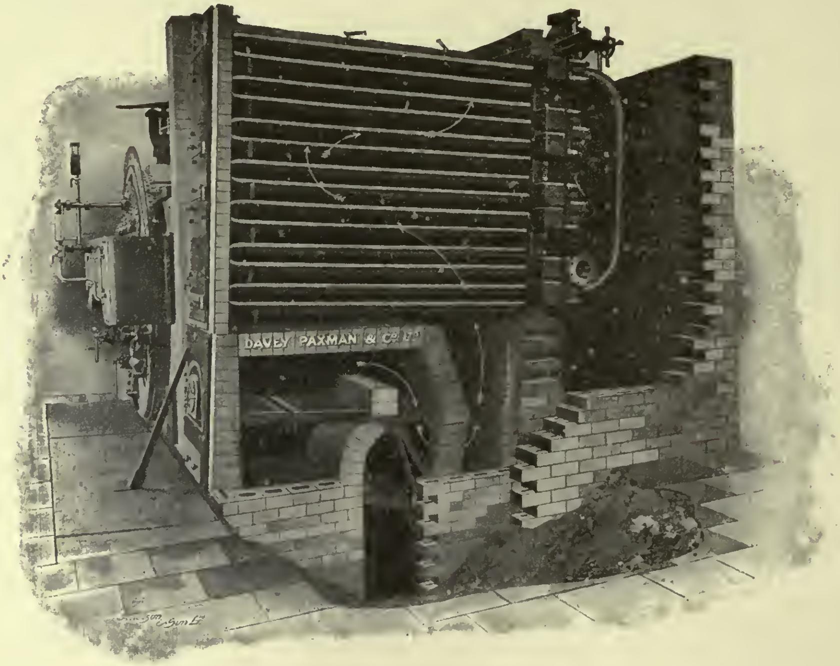
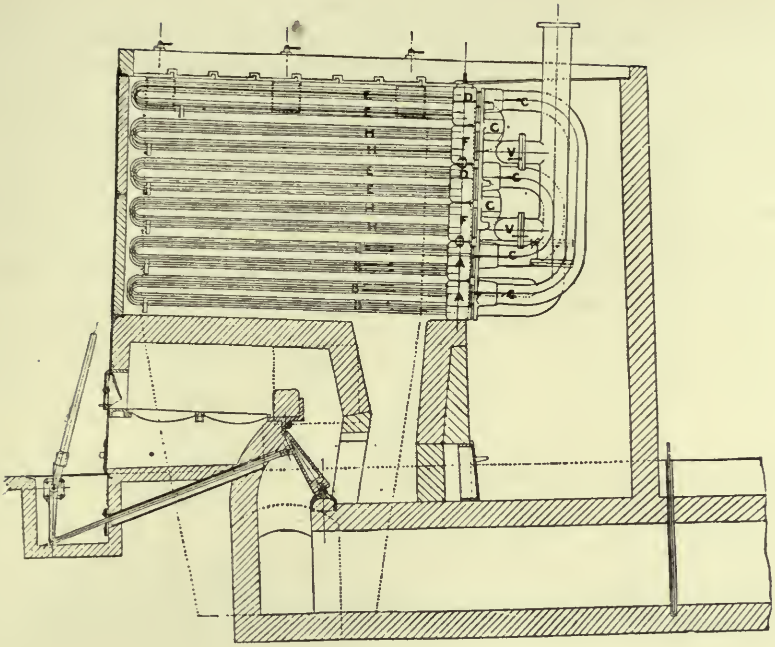


FIG. 214.—Davey-Paxman Superheater.

The course of the steam in the moderate temperature superheater is shown by arrows in Fig. 215, which will be easily understood by carefully following the sequence of the letters. The amount of superheat imparted to the saturated steam in this superheater is roughly one-third of that given in the high temperature superheater.


D
C
C
V
F
D
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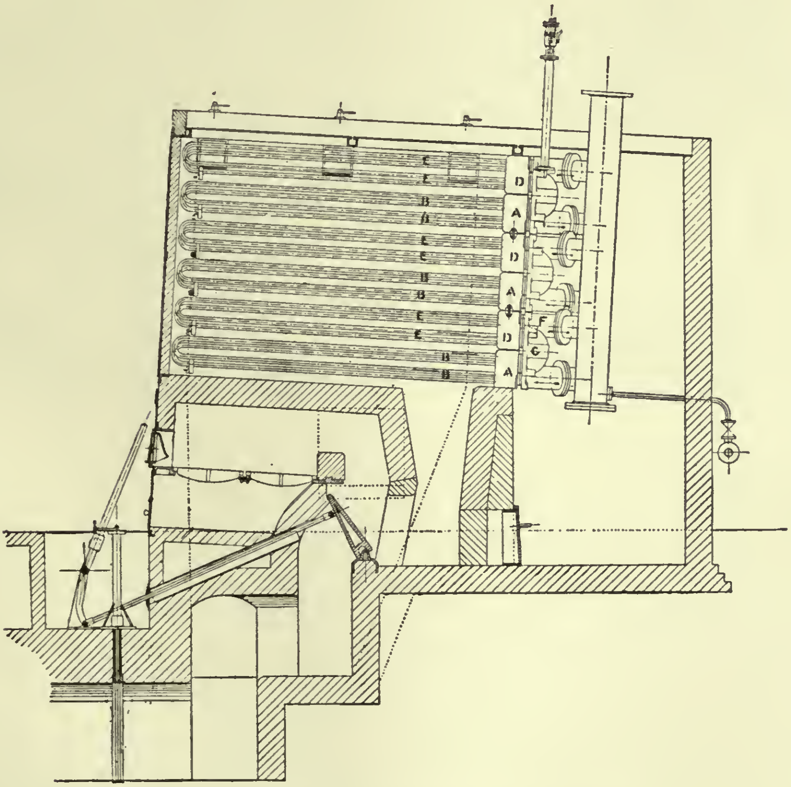
B
B

FIG. 215.


E
E
B
B
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B
B
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E
B
B
D
A
D
A
D
A
F
G

FIG. 216.—Sections of Davey-Paxman Superheaters.

The superheater equipment consists of furnace and front fittings, dampers, safety valve, thermometer, blow-out pipes and cocks, cover plates for the superheating and header chambers, firing tools, and other accessories necessary for the proper working of the superheater.

These superheaters can be arranged to fulfil effectively the following conditions:—(*a*) to vaporise the entrained moisture in saturated steam and superheat 20° to 50°, so that the steam delivered at engine stop valve is gaseous steam; (*b*) to vaporise moisture and superheat from 100° to 150° for increasing the economical performance of modern high-class engines, or to enable the steam to be conveyed considerable distances in a gaseous state; (*c*) to give the maximum superheat to steam for use in engines of special design, expressly adapted for giving the highest obtainable economy.

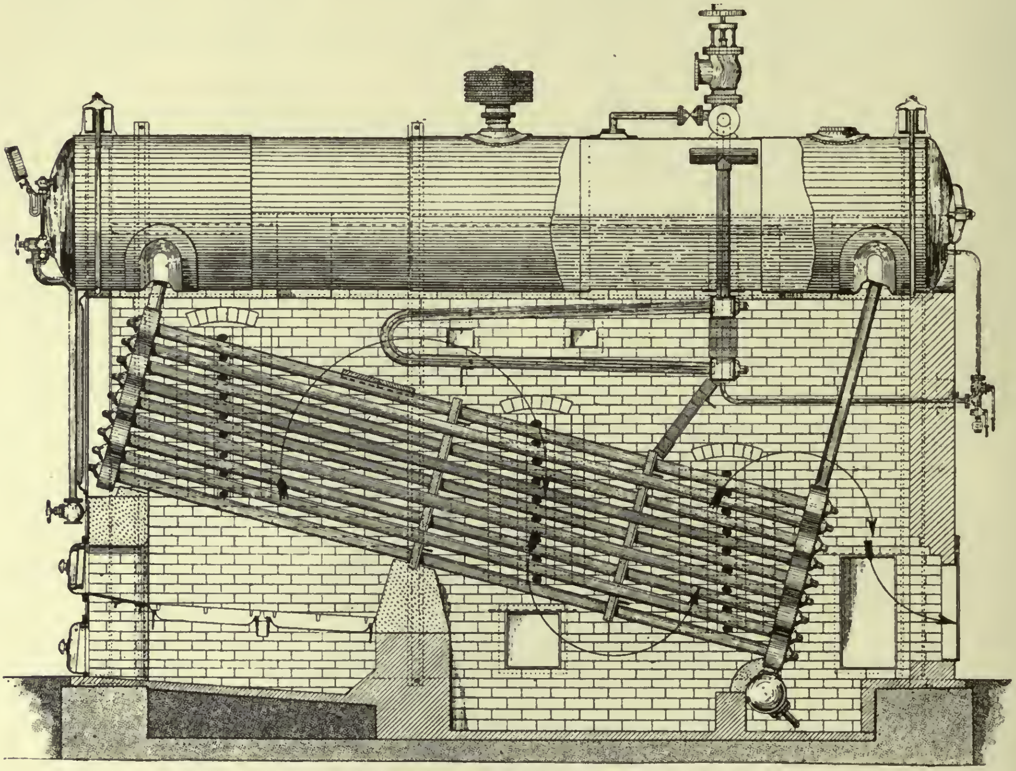


Fig. 217.—Babcock & Wilcox Superheater.

The superheaters are also suitable for heating large quantities of steam for sterilising fabrics and materials, and freeing them from filth and fats.

In a recent test carried out by inspecting engineers for one of the leading electric traction companies, the temperature of the gases in the cavity walls forming the down flues on a level with the fire-brick arch varied from 400° to 420° Fahr., when steam at 160 lbs. pressure was being superheated to 680° Fahr. The temperature of saturated steam at 160 lbs. pressure is 370°.8 Fahr., and the low temperature of the escaping gases is entirely due to the novel arrangement of the course of the steam through the superheater, in conjunction with the special form of foundation adopted, whereby the hot gases are trapped in the upper part of the superheating chamber.

Such, briefly, is the construction of the superheater and the course of the steam throughout. In addition to this, there are economical advantages obtained in the special arrangement of the superheater setting. The furnace is placed immediately underneath the superheating chamber, so that the radiation of heat from the fire-brick arch is directly transmitted to the gases surrounding the tubes. A bypass damper is provided, which can be thrown over to close the admission to the superheating chamber, when the whole products of combustion are short-circuited either to the flue of an adjacent boiler to produce steam or to the chimney. This arrangement admits of the fire being conditioned before the superheater is put to work; and the damper can be thrown over and the superheating of the steam commenced right away without any preliminary flooding of the superheater with water during the lighting of fires, which is a common arrangement with other types, and is open to grave objections on account of the deposit of scale on the tubes. Further, if through any mischance a superheater when charged with water is put into operation, a serious explosion may occur, owing to the potential energy of the heated water finding vent on the sudden removal of pressure; or, through carelessness, large bodies of water may be carried across into the engine without any adequate warning being given, with disastrous results to high-class plants.

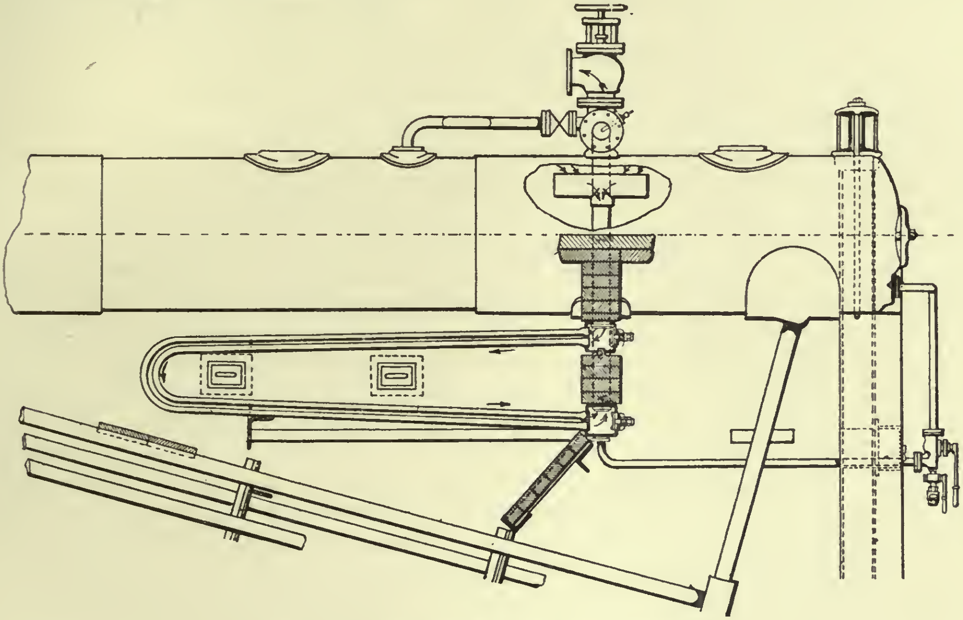


Fig. 218.—Babcock & Wilcox Superheater.

The superheating chamber is flanked on either side by cavity walls, which are connected by six small registers on the top of the superheater, so that the escaping gases are under full control, and by natural convection only the colder gases are drawn away to the chimney. Suitable access doors of the full width of the superheating chamber are provided in the front and rear of the superheater.

Very much akin to this reverberatory furnace system is that of placing a superheater in the boiler flues or combustion chamber, where only hot products of combustion can reach it.

The Babcock & Wilcox superheater belongs to this class, and the principles of its working are these. Fig. 217 shows the complete arrangement. Fig. 218 shows the details.

Firstly, the necessity of placing the superheater in a position where a suitable temperature is available. If a boiler is working under economical conditions the difference between the temperature of saturated steam and of the escaping flue gases leaving a boiler is not sufficient to superheat the steam produced, unless the surface of the superheater be very large; hence, generally speaking, a superheater should not be placed in the flue between boiler and chimney. On the other hand, if placed too near to the boiler furnace, possibly an excessive temperature may be attained which would cause damage.

Secondly, the superheater should be constructed to admit of all its parts expanding and contracting freely, without severe strains being put on any of the joints, which might cause them to leak ; flanged joints, when exposed to furnace gases, should be avoided wherever possible.

Thirdly, provision should be made to protect the superheater against overheating when steam is not passing through it sufficiently to carry away the heat supplied by the furnace gases.

This superheater is not subject to the immediate action of the fire, as the furnace gases must first pass through the front part of the boiler, which comprises a considerable heating surface. Assuming the boiler to be in regular work, and the firing even, no great fluctuations in temperature can take place where the superheater is fixed. Moreover, it is readily accessible for examination, and for the renewal of tubes.

There are no flanged joints ; all the tube joints are expanded.

Freedom for expansion is provided for by the tubes being free at one end, and by the manifolds not being rigidly connected with each other.

Prevention against overheating during steam raising is ensured by the arrangement for flooding with boiler water, and using the superheater as part of the boiler heating surface.

As will be seen, the tubes are bent into a U-shape, and connected at both ends with manifolds, one of which receives the saturated steam from the boiler, the other collecting the steam after it has traversed the superheated tubes, and delivering it to the valve, placed above the boiler.

The flooding arrangement consists merely in a connection with the water space of the boiler drum, and two cocks, by means of the larger of which, at will, the water enters the lower manifold and fills the superheater to the boiler water level. Any steam formed in the superheater tubes is returned into the boiler drum through the collecting pipes, which, when the superheater is at work, conveys saturated or natural steam into the upper manifold. Prior to opening the superheated stop valve and using superheated steam, the water is drained away from the manifolds by the flooding pipe and the smaller cock. A sight glass is attached to the drain outlet for observing that all the water has been drawn out of the superheater when draining.

The water in the boiler steam and water drum is, by reason of the fact that it is heated to steam temperature, deprived of most of its impurities, hence there is little fear, in flooding the superheater with the boiler water, of the tubes being choked by sediment. Sometimes the superheaters are placed in the boiler flues. It is well, wherever possible, to put the superheater in such a position that it will be acted upon by products of combustion at a temperature of 1000° Fahr., or higher. In order to accomplish this in the case, for example, of Lancashire boilers, superheaters are sometimes put into the downtake chamber at the rear end of the boiler. When this is done the tubes of the superheater inevitably become red-hot during steam raising, unless they are filled with water during this time.

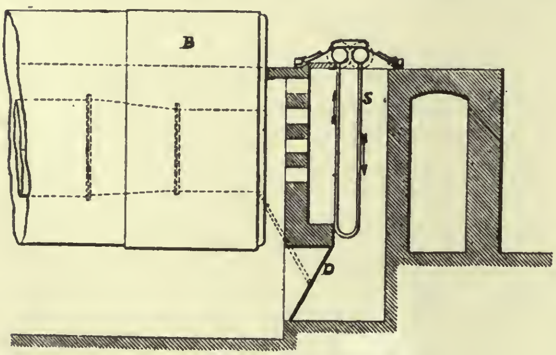


FIG. 219.—Watkinson Superheater.

Professor Watkinson, who has given much attention to this subject, with the object of obviating these difficulties, and in order that the degree to which the steam is superheated may be instantaneously regulated, devised the shunt-circuit system of supplying hot products of combustion to the superheater. The application of this system to a Lancashire boiler is shown in Fig. 219. B is the boiler, S the superheater, and D the regulating damper. During steam raising, and whenever the engines are standing, the damper is closed, and the products of combustion all flow downwards to the bottom flue. When the damper is open some of the products of combustion flow through the superheater chamber, and by adjusting the position of the damper the degree to which the steam is superheated can be instantly regulated. The damper, being in a comparatively cool position, does not burn away. This system works exceedingly well, and the saving in coal effected by it averages about 20 per cent.

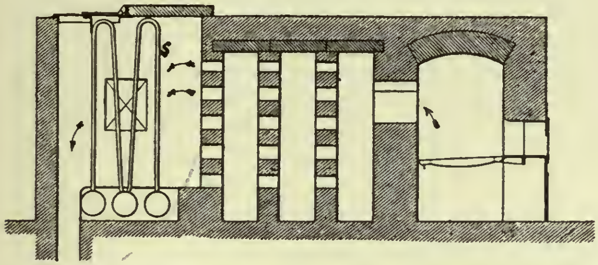


FIG. 220.—Watkinson Superheater.

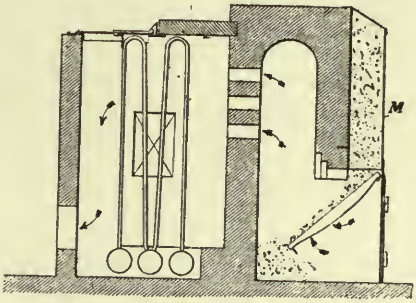


FIG. 221.—Superheater.

Fig. 220 illustrates an independently fired superheater combined with an ordinary furnace. When an ordinary furnace is used in conjunction with a superheater the furnace must be very frequently attended to, or great fluctuations in the temperature to which the steam is superheated will occur, due to variations in the condition of the fire. There is also danger of the superheater acting as a condenser whenever the furnace door is opened. In order to overcome these difficulties the author provides a heat reservoir, in the form of perforated brick walls, as shown. This arrangement gives very satisfactory results, but it is not as easily controlled as the gas producer type, and it requires more attention.

An intermediate type is illustrated in Fig. 221. In this case ordinary fire-bars are used, but a magazine M is provided, into which the fuel is fed at the top, and from which it automatically slides down to the fire-grate. A water trough, not shown, is provided in the ashpit.

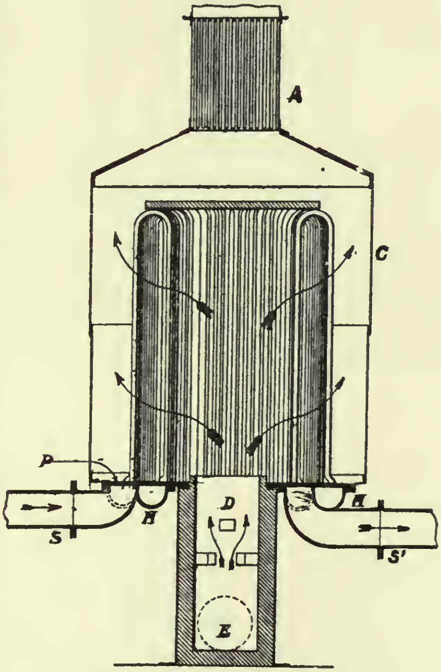


FIG. 222.—Watkinson Superheater.

For marine, locomotive, and some other purposes, where the space available is very limited, and where it is undesirable to have a large mass of brickwork, the arrangement of superheater shown in Fig. 222 is preferred. The tubes are of the inverted U-form, and they are expanded into an annular tube-plate P, and this tube-plate is bolted to the header H, which is either a steel casting or a forging. The steam enters the outer compartment of the header by the pipe S, and leaves the inner compartment of the header by the pipe S^1. A light steel casing C surrounds the superheater, and the products of combustion flow across the tubes, as indicated by the long arrows. The superheater is shown as arranged for the use of oil fuel, but it can be fired with any kind of fuel burned in an ordinary furnace or in a gas producer. An economiser of the air-heater type is shown at *a*, and by means of this the efficiency of the plant is appreciably increased. The outside diameter of the casing for a superheater of this type for 150,000 lbs. of steam per hour is about 11 feet.

The third system of firing is a development of the second, namely, by a regular gas producer and flame combustion of the gas. The gas producer is a direct descendant of the old reverberatory furnace, in which the fuel is laid deep on the fire and the air supply is restricted, so that combustion goes on slowly at the bottom, creating sufficient heat to distil the gases from the fuel in the upper layers ; and to form carbon mon-oxide in the hot under layers, the gases are conducted from the top to the combustion chamber or furnace where they are required for use.

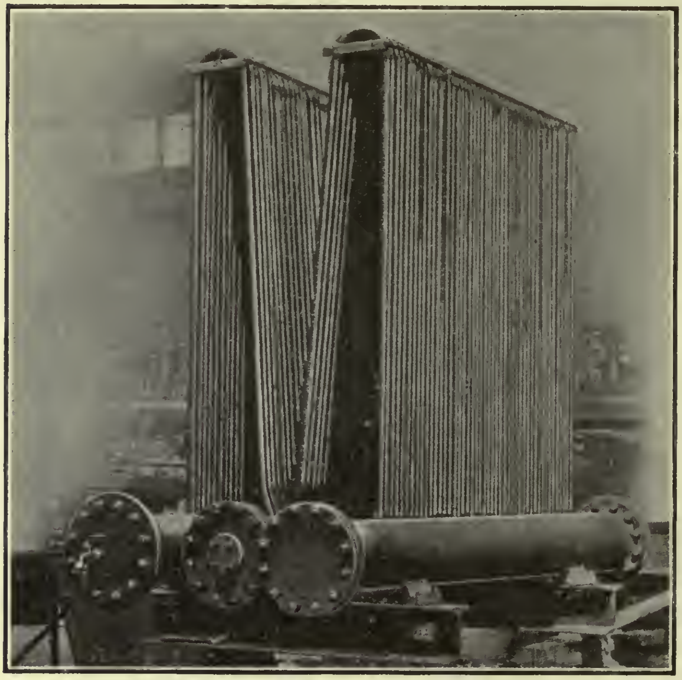


FIG. 223.—Watkinson Superheater.

Professor Watkinson has developed these producer gas heaters for superheating to a high degree of perfection.

The general construction of Professor Watkinson's independently fired type of superheater is illustrated in Fig. 223, which is from a photograph taken during construction in the works of the makers, Messrs. Mechan & Sons, Scotstoun Iron Works, Glasgow.

The headers are of wrought iron or mild steel. The tubes, which are of solid-drawn steel, are arranged in rows at very small distances apart, so that the products of combustion in passing between the tubes are divided into thin sheets, and in this way they are compelled to give up their heat much more readily to the tubes. The action of tubes arranged in this way may be likened to the action of the wire gauze in a Davy safety lamp. The products of combustion flow transversely to the tubes, and the efficiency of the arrangement is well illustrated by the fact that, although the gases are white-hot as they approached the first wall of tubes, they pass away from the superheater at a temperature of about 460° Fahr.

The high rate of heat absorption, obtained as described above, enables the size, weight, and cost of the superheater to be greatly reduced as compared with any other arrangement. The close spacing between the tubes is rendered possible by the tube ends being drawn down to a smaller diameter for expansion into the headers. The same object may be accomplished by suitably bending the tubes near their ends, so as to allow of the tube holes in the headers being zigzagged. The tubes are expanded into

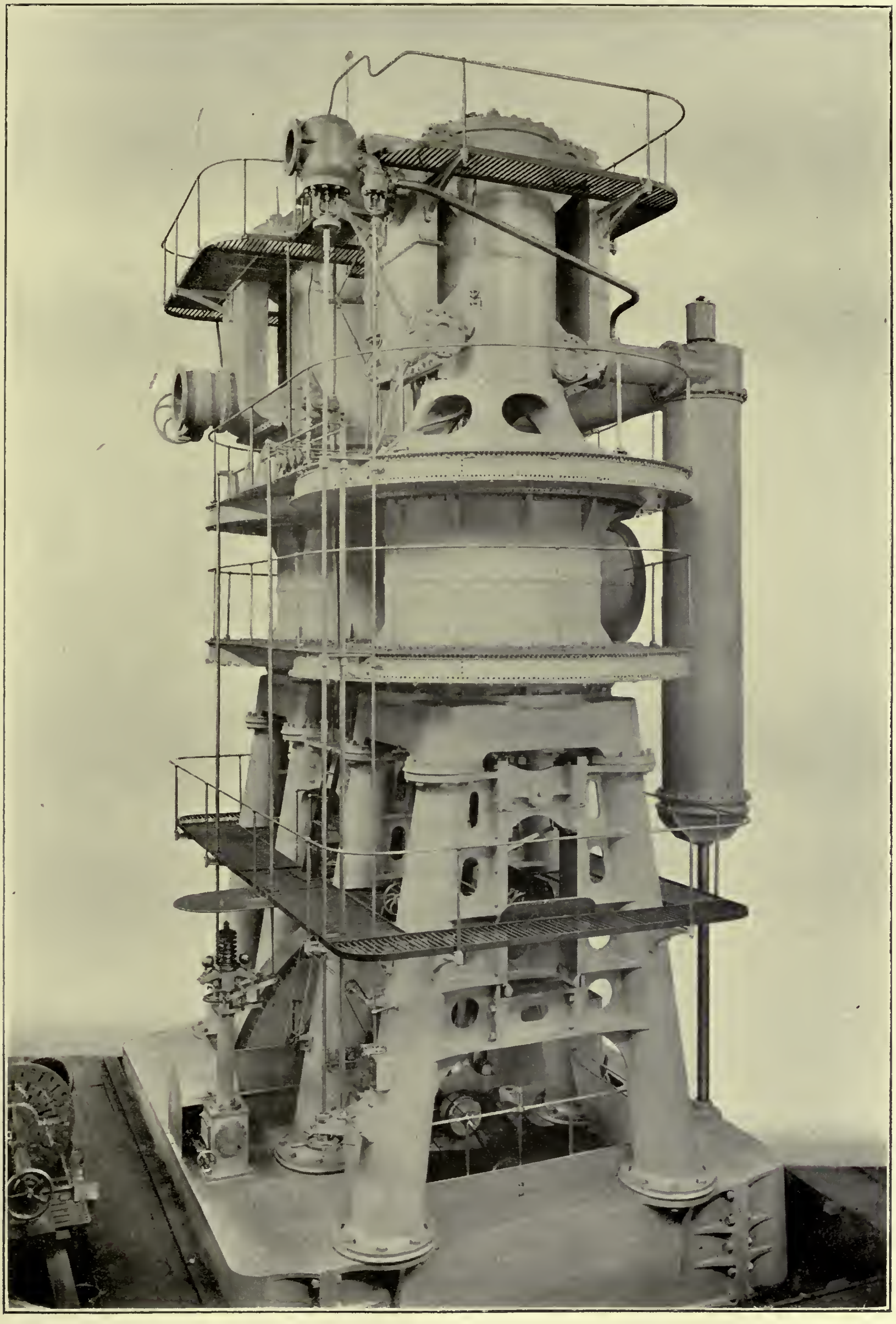


Plate IV.—Modern Compound Vertical Blowing Steam Engine (Richardsons, Westgarth, & Co., Middlesbrough).

the headers from the ends, by means of bevel-wheel driven expanders. Any tube can be put in or taken out without interfering with any other tube. As the tubes are nearly vertical, soot and ash do not accumulate on them, and all the tubes can be perfectly drained.

The cost of construction might be reduced by dispensing with the middle header and making the tubes of the inverted W-form instead of the inverted U-form. With the inverted W-form of tube, however, complete drainage of the tubes is impossible, and blowing through with steam does not appreciably help matters, as already explained.

The superheater illustrated in Fig. 224 is being erected at a colliery in Lanarkshire. At present nine boilers are required to supply the steam for the various engines, but it is expected that only six will be required when the superheater is started. The fact that it pays to adopt superheating even at a colliery, where the cost of coal is a minimum, indicates that the saving in other cases, and especially on board ship, will give a very handsome return for the money invested in superheating appliances.

Fig. 224 is a longitudinal section of this superheater and its furnace, which is of the gas producer type. A superheater of this type has been at work night and day for over two years, and a recent examination of it showed that the tubes were apparently as perfect as when new. In this case six boilers were in use before the adoption of the superheater, whereas now only four are required.

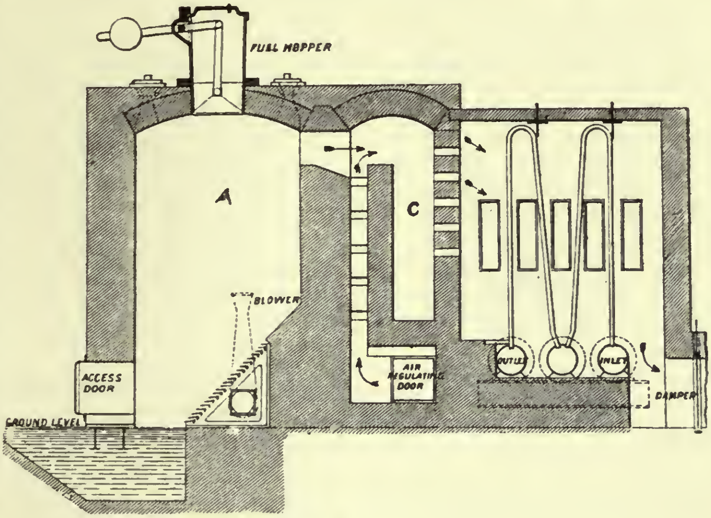


FIG. 224.—Watkinson Superheater.

Wherever possible the gas producer type of furnace in conjunction with independently fired superheaters is used, because with this arrangement the attention required is less than with an ordinary furnace, the fuel being fed in through the hopper, say, once an hour, and the ashes being removed at the bottom of the producer every twelve hours. Another advantage is that smoke is entirely got rid of; but the greatest advantage is due to the perfect control which is thus obtained of the degree to which the steam is superheated.

In Fig. 224 A is the gas producer. The air for primary combustion is blown in through the special grate shown near the bottom of the producer. The air for secondary combustion enters at B, and passes up, as indicated by the arrows, through ducts provided between the gas producer A and the combustion chamber C. The products of combustion flow from the combustion chamber to the superheater through the perforated wall shown.

The degree to which the steam is superheated can be almost instantaneously regulated by varying the rate of supply of steam to the blower. By means of a very simple attachment the rate of supply of steam to the blower can be controlled automatically so as to maintain the temperature to which the steam is superheated constant.

The Watkinson independently fired superheaters described require much less attention than steam boilers, and it is probable that their life will be at least as long as that of boilers. As, by the adoption of this complete system, the amount of steam required by the best triple-expansion engines can be reduced to two-thirds of that required with saturated steam, the number of boilers required will also be reduced to two-thirds. The saving in cost on these will more than pay the cost of the superheaters, and the weight and the space occupied by the superheaters are very much less than the weight and space occupied by the boilers displaced.

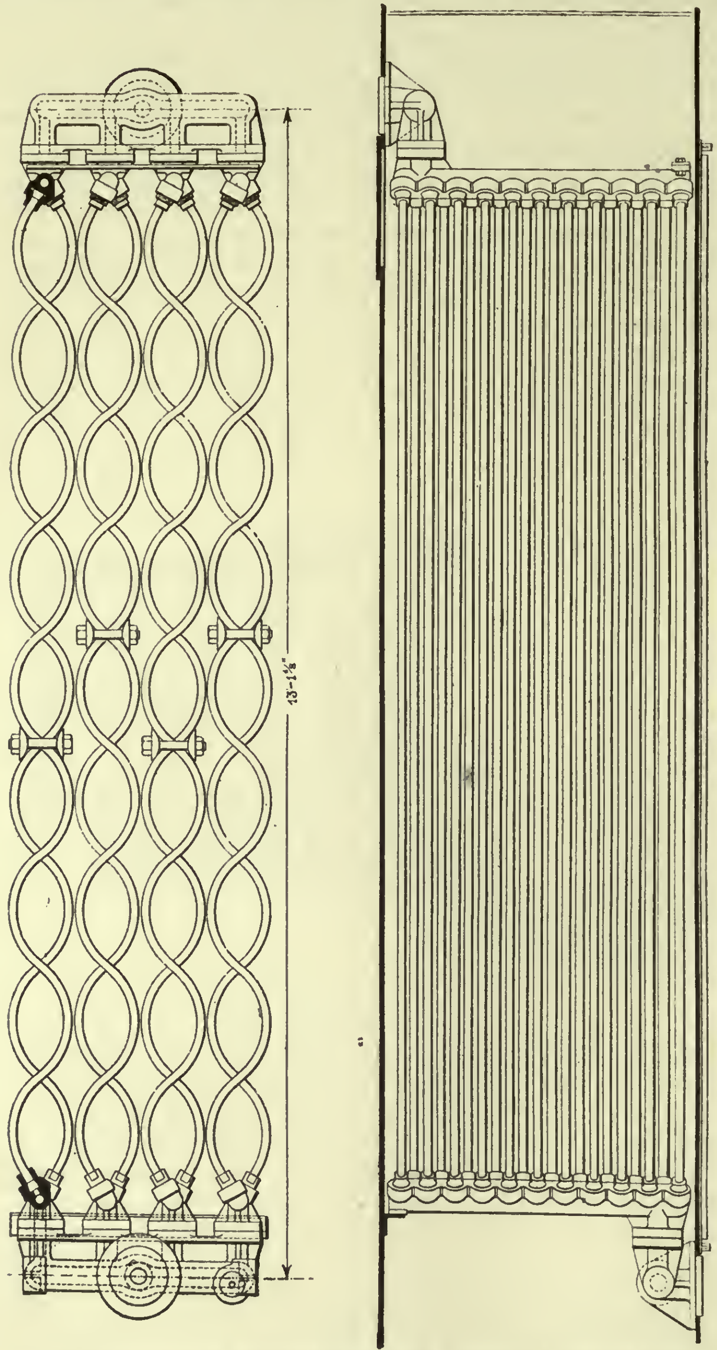


Fig. 225.—Central Marine Engineering Company's Superheaters.

For cases where only one or two boilers are used it is generally best to heat the superheater with products of combustion from the boiler furnace; but for larger installations it is better to use a separately fired superheater. One reason for this is, that if a superheater is applied to each of a battery of boilers, it is not known what each one is doing. One of the superheaters may be applying nearly red-hot steam, whereas another, the boiler of which is being lightly fired, may be acting as a condenser. The cost of a separately fired superheater, and of the pipes and bypass valves, is also less than the corresponding cost where a superheater is applied to each boiler. The greatest advantages, however, of the separately fired superheater in a large installation are the greater certainty of action and the greater ease of control of the degree to which the steam is superheated. The combination of a gas producer with a superheater, described, overcomes all difficulties connected with this type of superheater.

Owing to the resistance offered by the superheater to the flow of steam through it there is necessarily a slight drop in pressure between the inlet and the outlet ends;

the amount of this drop in pressure depending on the rate of flow of the steam through the superheater. The allowable drop in pressure in any given case depends on the conditions of the case, and it should be definitely determined beforehand to suit these conditions. In general the drop in pressure should not exceed, say, 3 lbs.

CENTRAL MARINE ENGINEERING COMPANY'S SUPERHEATER

The Central Marine Engineering Company's superheater has been successfully applied to steamships, where it is heated by the flue gases.

It consists of three divisions, in each of which are two cast-steel headers (Fig. 225). Between these are 168 waved tubes, through which the steam passes on its way from the boiler to the engine, entering at the top and leaving at the bottom. The waved form of the tubes ensures that the steam shall be continually agitated, and that all parts shall come in contact with the hot metal. The lower part of the waved pipes is naturally hotter than the upper part, for the gases from the boiler strike here, and hence the steam is heated progressively, while the gases are cooled in the same manner. When they leave the top of the superheater they flow up through nests of vertical pipes to the fan, by which they are aspirated and forced into the funnel. These vertical pipes heat the air which is drawn into the furnaces by the fans, the air entering through openings in the casings, and flowing downwards among the pipes, and thence to the ashpits.

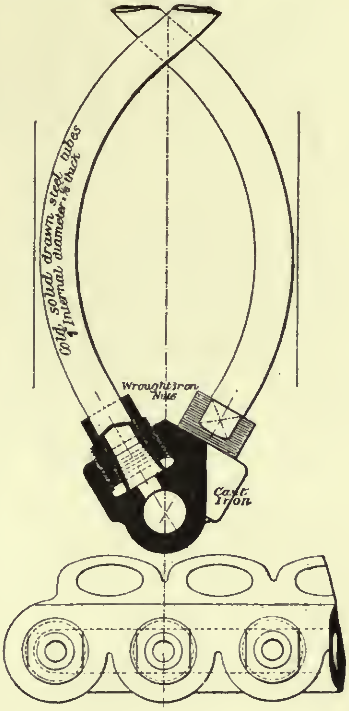


Fig. 226.—Central Superheater.

Fig. 226 illustrates the method of securing the superheater tubes into the headers; a more recent arrangement still is made to suit natural draught conditions, in which the superheater is placed in the uptake of the chimney, with damper and other arrangements fitted to cause the hot waste gases either to pass through the superheater chamber or directly into the funnel at will.

Recent information from the owners of the "Inch" steamers that the *Inchmarlo*, making a voyage of 380 days' duration to all parts of the world, and taking in coal just as they could find it at the various ports of call, has given the following results:—Dead weight carried, 5700 tons; average speed over the whole time, 9 knots; coal consumption, 15¾ tons per day. This is a record in economical coal consumption that has never been previously touched in marine work. It works out to the merest fraction over 1 lb. of coal per indicated horse-power per hour (1.001 lb. per indicated horse-power). These superheaters will be referred to again in connection with marine boilers.

A design for a locomotive superheater has been proposed and illustrated by Mr. John Reekie, in a discussion on superheaters before the Institution of Engineers and Shipbuilders in Scotland. Fig. 227 shows the general idea of the design. The superheating tubes are carried through a corrugated flue in the central part of the tube plates; a damper at the fire box end on the butterfly valve principle is used to adjust the amount of superheat.

There can be no doubt superheating, if scientifically carried out, pays in every case. It is one of the economies upon which the steam engine relies for its continued existence as a prime mover. The internal combustion engine competition will compel the adoption of superheating to keep the steam engine to the fore.

The gas engine itself requires no increase of temperature at the upper end of its heat range, but it suffers from its inability to utilise the heat at the lower end. The exhaust escapes at about 1200° Fahr. temperature, and pressures between 25 and 50 lbs. per square inch. This is at present an unavoidable loss, for the highly heated gases cannot be expanded beyond the volume in the cylinder which they occupied when cold at the end of the suction stroke. Were it not for this great loss in the internal combustion engine exhaust, the steam engine, even with high vacuum at one end and superheat at the other end, would make a poor comparison.

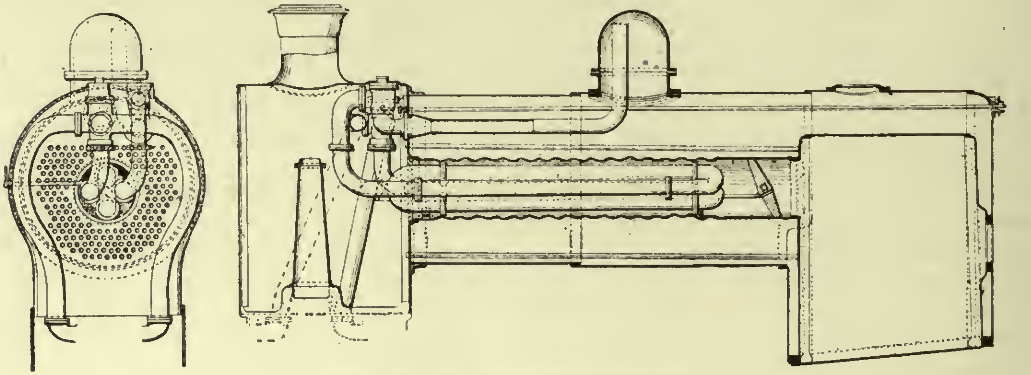


Fig. 227.—Locomotive Superheater.

Both superheating and high vacuum entail considerable additional plant and attendance, but the steam engineer does not hesitate to adopt them, even if the saving is only a small one; whereas at the present stage of internal combustion engine development, engineers are more concerned with the details of mechanism than with economy, so that no attempts are made to add additional apparatus for utilising the heat to a greater extent.

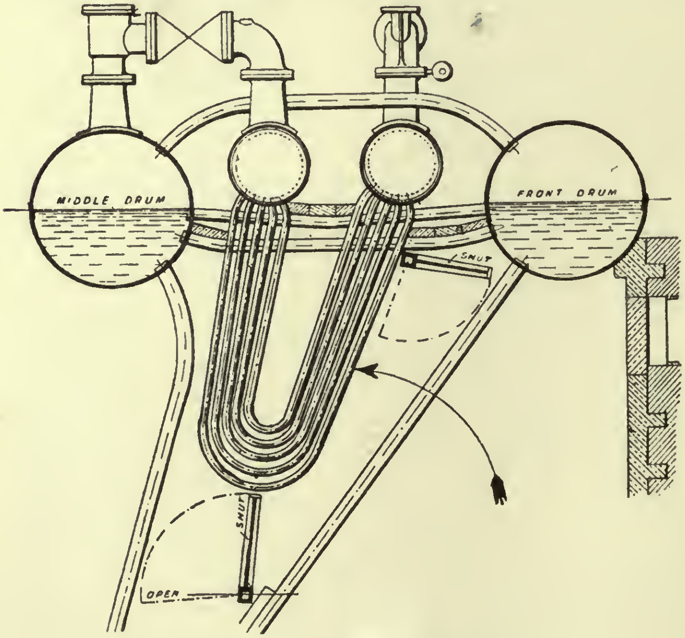


Fig. 228.—Stirling Boiler Superheater.

The compounding of gas and oil engines has proved a difficult problem. The only successful application of the waste heat in the exhaust gases has been to pass them into a steam boiler flues to raise steam to be used for power and other purposes.

The Stirling boiler superheater is shown in Fig. 228 by the arrow on their water tube boiler. It is a high temperature superheater, the inlet and outlet headers or drums of which were of sufficient size to enable a man to have access to the interior. Divisions were arranged in the drums to give a long travel and rapid circulation to the steam in the tubes. The path of the flue gases was shown by the arrow; the first bank of main tubes in the boiler intervened between the superheater tubes and the furnace. When the superheater was out of action the usual water flooding appliances were brought into operation to prevent rapid deterioration of the tubes. Temperatures up to 750° Fahr. could be obtained with this superheater. With the low temperature superheater, which had straight horizontal tubes, a temperature of only about 500° Fahr. was obtained, as it was arranged in the Stirling boiler somewhat remote from the fire, between the middle and back drums, and received the action of the furnace gases only after the latter had travelled over the first three banks of tubes.

BELLISS AND MORCOM'S ENGINES, WITH SUPERHEATING

The thermal efficiency of an engine is increased theoretically and actually by increasing the range of temperature of the working fluid. This may be done by increasing the boiler pressure, by compounding, by superheating, and by lowering the back pressure. Superheating has many advantages, and the subject is one which up to the present does not appear to have had from British engineers the attention it deserves. The cause of this is not easy to explain. The chief obstacles in the past have been that of internal lubrication and the life of the superheater, but there has been in this country a good deal of conservatism amongst engineers, not to say prejudiced hostility, to its use. It must not be forgotten that Hirn, as long ago as 1857, and others have pointed out the great advantages of superheating, and this advice has been acted upon by German engineers, and in a lesser degree in America, with far more energy and earnestness than in this country. The difficulty of lubrication consequent on the use of superheated steam is in general confined to the high pressure slide valve and cylinder, and as the requisite materials are now not difficult to obtain it is consequently all the more surprising that the practice has not made further headway in Britain.

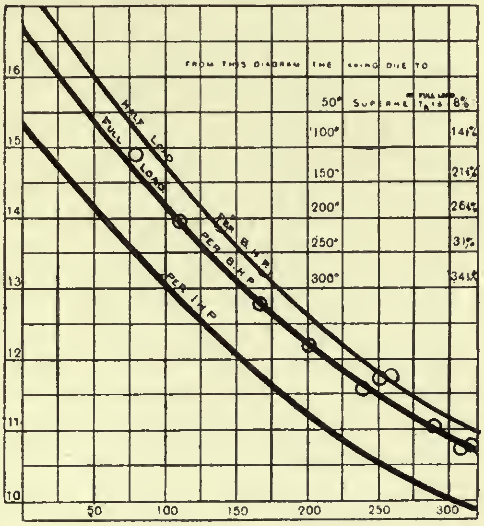


Fig. 229.—Diagram illustrating the effect of Superheat on Steam Consumption. Being Full Load Condensing Results from 300 Horse-Power "Belliss" Triple-Expansion Engine.

The loss of energy due to the use of steam initially saturated when entering the cylinder is of course caused by condensation. This takes place at two periods of the stroke,—first, on entry, due to the steam coming in contact with the cylinder walls cooled by the previous expansion and exhaust; secondly, during expansion, owing to the temperature of the steam falling more rapidly than the saturation temperature. Some re-evaporation does take place at release, as the pressure falls below that corresponding to the saturated state, but this only causes a further abstraction of heat from the cylinder walls, and chills them further to receive the next volume of incoming steam.

Superheating, then, may have as its object the removal of the first or of both of these losses due to condensation. Some people have an idea that only small superheats are of practical value for increasing the thermal efficiency of an engine, but this is only partially true. By increasing the amount of superheat up to the point necessary to obtain dryness at the point of cut-off the thermal efficiency also rapidly increases. Beyond this point increase in superheat is attended by increase in thermal efficiency, but at a somewhat reduced rate. Now, in order to obtain dry steam at *cut-off only*, a superheat of from 150° to 250° Fahr. is required, depending chiefly upon the number of expansions employed. A further superheat of at least 100° Fahr. is necessary to secure dryness at release. In compound engines it is practically impossible to sufficiently superheat the steam before entry to the high-pressure cylinder to enable it to continue dry up to the point of cut-off in the second cylinder. To obtain the advantages of superheating in the second and later cylinders, the practice of reheating is sometimes resorted to, and is extensively in vogue among American engineers, and their claim that considerable economy results therefrom is apparently due to the fact that they do not use absolute measures for work performed, but indicator diagrams thereof. It is not a profitable arrangement to reheat steam for the second and later cylinders; quite as good economy can be obtained with the same quantity of heat supplied in the form of initial superheat. The only possible object of going to the additional expense of adding a reheater is that thereby the engine may be able to work with an initially reduced temperature of steam. The Belliss engine, however, experiences no difficulty whatever in working with the highest temperature it can be given, and there is ample experience to show that the effect of superheated steam on these engines is attended by great economy and without harm in any way.

EXPERIENCE WITH SUPERHEAT

Environment usually determines whether a flue-fired or independently fired superheater is the better, but in designing a new station, independently fired superheaters are preferable, and proportioned to give a temperature at the engine stop valve of 500° to 550° Fahr., their use ensuring a constant temperature, and also reduction in boiler capacity. With a flue-fired superheater the temperature varies considerably, rendering it inexpedient to reduce the boiler capacity in so great a degree.

It is interesting to have the actual experience of a responsible practical engineer upon this superheating question, as it was found in practice by Mr. A. S. Biggart, whose account of an actual case was given in a discussion on superheaters before the Institute of Shipbuilders and Engineers in Scotland. A few years ago his firm laid down a large new work, and, after considering the question, decided to put down a generating station that would ultimately be a centre of power for their old as well as their new works. The boiler plant adopted was of the Babcock & Wilcox type, without superheaters. For some time after starting practically no trouble was experienced with the boilers or engines, but by and by, as the power demanded of the boilers became greater, trouble in the engine-room began. That was entirely due, so far as he could judge, to the saturated state of the steam. As power was added it gradually got worse, till the engines could not be run without having all the drain cocks at least partially open. On asking the representatives of Messrs. Babcock & Wilcox to look into the matter they reported that the boilers were being overtaxed, and that if the boilers gave off dry steam instead of saturated steam there should be abundance for the power that was then required from the engines. Then arose the question of further additions to the boiler plant. After looking into the matter, and as further power was about to be taken from the station, it was decided to add another boiler plant with superheaters, thus doing away with saturated steam. This had been done, and the new plant had been working for some time. The station was now worked wholly with superheated steam, using only the new boiler plant, which was an exact duplicate, as far as grate area and heating surface was concerned, of the old plant. All the old engines in the engine-room, besides additional plant, would now be driven with superheated steam. The quantity of water evaporated in the boilers by the new plant was practically as six to nine in the old plant, in spite of the additional power used. In short, it came to this; had superheaters been added to the old plant they would not have required to add any new boiler plant at all. That was rather a remarkable experience. For practical purposes the results were convincing. Secondary considerations, in exceptional cases, came into play in working with superheated steam; it might be said with confidence that even 40 per cent. saving was not unknown—not due to superheating directly, but due to being able to shut drain cocks, and other savings in the engine and outside of it incidental to superheating.

THE PIELOCK LOCOMOTIVE SUPERHEATER

With this superheater fitted to a 4-cylinder compound locomotive at the St. Louis Exhibition, 1904, it is claimed that a saving of 15 to 18 per cent. of coal has been effected, and 20 per cent. of water. It has the merit of great simplicity, can be fitted to old or new locomotives, and is not exposed to overheating. As can be seen on referring to Figs. 230 and 231, it is of a very simple form, being merely a small steel box surrounding part of the tubes, and situated near the centre of the boiler length. The pressure inside and outside the box is the same, and it is only necessary to expand the tubes passing through it sufficiently tight to prevent leakage, this being done with a special mandrel made for the purpose. The weight of the box is supported by the tubes, no fastening or connection to the boiler being required. A drain cock, coming out through the bottom of the boiler, is fitted to the superheater box, and by its means leakage can readily be discovered. The boiler tubes are made slightly taper from the smoke box end to the fire box, so as to admit of removal without difficulty. The box is divided into several connected compartments by vertical plates arranged parallel to the tubes. Steam enters the superheater through the pipes E_1 and E_2, Figs. 230 and 231, and is conducted round the tubes in the direction indicated by the arrows to the pipe A, finally arriving at the steam chamber enclosing the regulator, where it remains until the opening of the latter allows it to find its way to the cylinders. A thermometer indicating the temperature in the steam chamber of the superheater is attached to the dome, and can be readily seen by the driver when standing on the foot-plate. The degree of superheat usually arranged for and in use on this engine is about 300° C. The amount of superheat required is regulated when designing by the situation and length of the superheater box. Of course, the nearer the superheater is placed to the fire box the greater the amount of superheat obtained; its position, however, is limited, for when very near the fire box there is great danger of burning out the tubes.

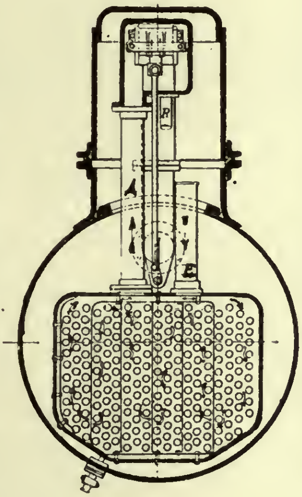


FIG. 230.

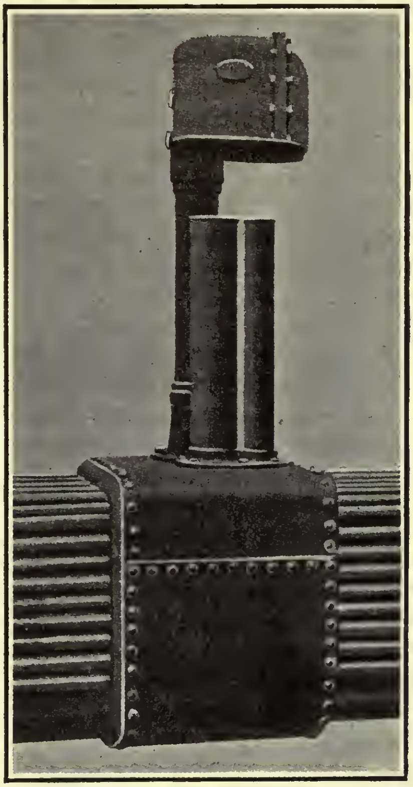


FIG. 231.

Pielock Locomotive Superheater.

THE SCHMIDT SUPERHEATER APPLIED TO LOCOMOTIVES

The Schmidt superheater is well known in land installations. It has been applied to locomotives for the Cape Government railways, built by the North British Locomotive Company at Hyde Park Works, Glasgow, from the designs of the Cape Railway's chief locomotive superintendent, Mr. H. W. Beatty, C.M.G.

The superheater is contained in the smoke box, and consists essentially of a series of solid-drawn steel tubes, 61 in number (see Fig. 233). These tubes are bent into concentric rings and arranged in vertical planes about an inch apart, each plane containing alternately two rings and one ring, the one ring being of a diameter intermediate between that of the other two. The intermediate and outer series of rings are arranged to come below the level of the bottom of the flue, the inner series not. The first nine rings, however, nearest the tubeplate are arched in decreasing radii from the front of the flue, thus forming a sort of chamber between the inner and two outer rings, into which the gases from the flue first flow. The inner series of concentric rings consist of 21 tubes, $\frac{5}{32}$ inch thick and $1\frac{5}{8}$ inch in outside diameter, while the intermediate and outer series consist of 19 and 21 respectively, $\frac{5}{32}$ inch thick and $1\frac{1}{2}$ inch in outside diameter.

These tubes are bent up at the ends and expanded into two boxes, or steam chests, attached to the smoke box—one to the right, the other to the left of the chimney (see Fig. 233). These boxes, which are made of cast steel, are about 4 feet long and almost semicircular in cross section, such a section being better able to withstand the high pressure to which it will be subjected than one rectangular in form. They are fitted with covers outside the smoke box, which, in the first instance, enables the tubes to be readily expanded, but which will be also convenient for purposes of examination. Steel casings attached by screws and distance pieces are placed over them, the space between being filled with asbestos to prevent radiation. The right-hand steam chest is divided into two chambers by a central partition wall, half the superheater tubes being expanded into one of the chambers, the other half into the other chamber, thus dividing the tubes into

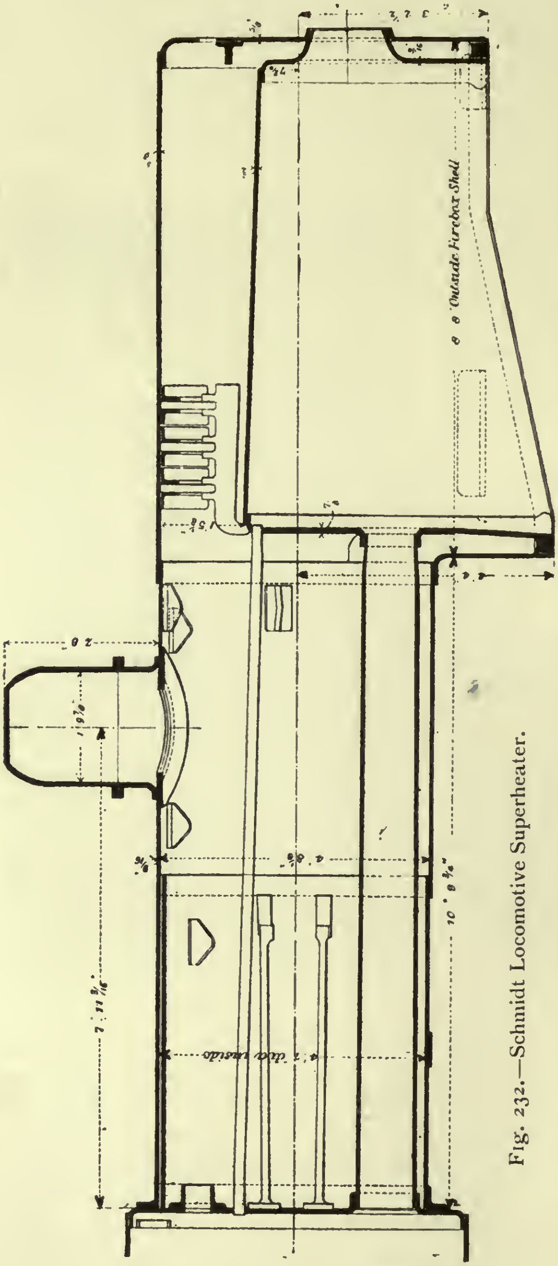


Fig. 232.—Schmidt Locomotive Superheater.

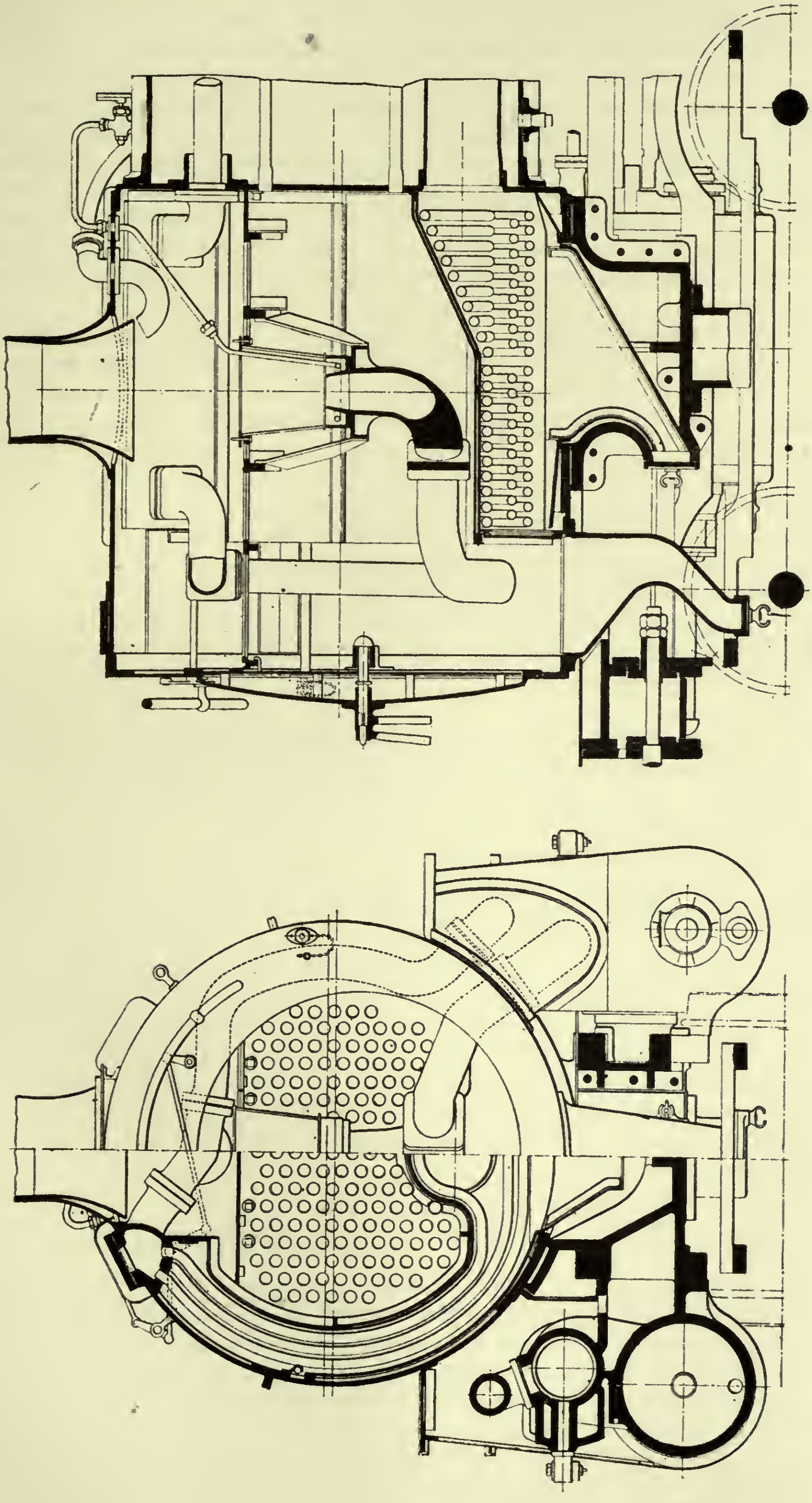


Fig. 233.—Schmidt Superheater.

two groups. This steam chest is connected at the tube-plate end to the regulator pipe, the other end being connected through the steam pipes to the cylinders. The left-hand steam chest has no partition, but it is merely a plain box open from end to end.

The superheater tubes are encased in a box, built up of thin steel plates adapted to the shape of the outer series of tubes (Fig. 233), and ending at the steam chests in two narrow vertical dampers, the whole being composed of several portions bolted together, and designed in such a way that any portion can be readily detached from the others should it be found necessary to examine the tubes. For further purposes of examination portions of the smoke box, extending from the bottom of each steam chest to about 9 inches above the centre of the boiler, and also the portion between the steam chests, can be readily removed.

The whole superheater occupies about two-thirds of the total length of the smoke box, the remaining third being occupied by the steam pipes (Fig. 232), which are placed in front of the superheater, and the exhaust pipes, which afterwards bend in some distance above the superheater, and end in the blast pipe. A shoot is provided below these pipes to collect the ashes coming through the boiler tubes, and an additional one below the superheater to collect those coming through the flue, thus preventing the choking of the superheater. Two perforated pipes, the perforations being at such angles as to sweep the whole of the tubes, are attached to the inside of the smoke box plate—one on the right and the other on the left. It is intended that steam or compressed air should occasionally be blown through them to remove the soot which has collected on the tubes. To this end the pipes project through the front of the smoke box and end in screwed caps, which can be removed when the engine is in the sheds and steam or compressed air applied. It will, of course, be better to use the latter if available, to avoid the risk of rusting the tubes.

The action of the superheater is quite simple. On opening the regulator, saturated steam is admitted to the rear chamber of the right-hand steam chest, and from there it passes through the tubes, arriving in a slightly superheated condition at the other steam chest. This steam chest, having no partition, permits the steam to return through the other half of the tubes to the front chamber of the right-hand steam chest, and from thence in a highly superheated condition it passes to the cylinders. During the passage of the steam through the tubes part of the gases produced in the fire box pass through the flue into the superheater, and, owing to the vacuum produced in the smoke box when the engine is running, circulate freely among the tubes, giving up their heat to the steam as it passes through, and finally rising to the dampers, and joining the gases from the boiler tubes, they pass away together through the chimney. The rate of flow of the gases can be increased or diminished by the driver regulating the dampers placed at the top of the superheater box. These dampers, as a protection to the superheater when the regulator is shut and steam is consequently not passing through the tubes, are connected to the lever actuating the blower in such a manner that they are automatically closed when the blower is put on.

THE WILLANS ENGINE AND SUPERHEATED STEAM

The Willans engine during the past few years has been extensively used for working with superheated steam, but the manufacturers have hitherto abstained from expressing any decided opinion as to whether the advantages outweighed the disadvantages until they had had an opportunity of satisfying themselves as to the conditions of the working parts of their engines after continued use at the higher temperatures. The result of their experience in this direction having been entirely satisfactory, they have placed at the disposal of steam users a few facts relating to the use of superheated steam in the Willans engine.

Steam superheated to a high degree undoubtedly gives greater economy than steam superheated to a moderate degree only; in fact, the principal reason for the recommendation of small degrees of superheat would seem to be merely the unsuitability of many types of engine for working above any but the most moderate temperatures.

But the real advantages of high superheat are brought out in the diagrams, where the gain from the use of superheat, under different conditions, is shown graphically. These diagrams are the result of careful experiments made by Messrs. Willans & Robinson, and show not only the saving in steam used by the engine, but also the saving in fuel. It is often said that though the use of high superheat may show a large saving in steam consumed, yet the real saving, that in fuel, is largely imaginary. The diagrams (Figs. 234 and 235) entirely refute this idea; the fuel saving, though less than the steam saving, is a very real one, and the engine which can work successfully with high superheat is really, as well as apparently, the most economical.

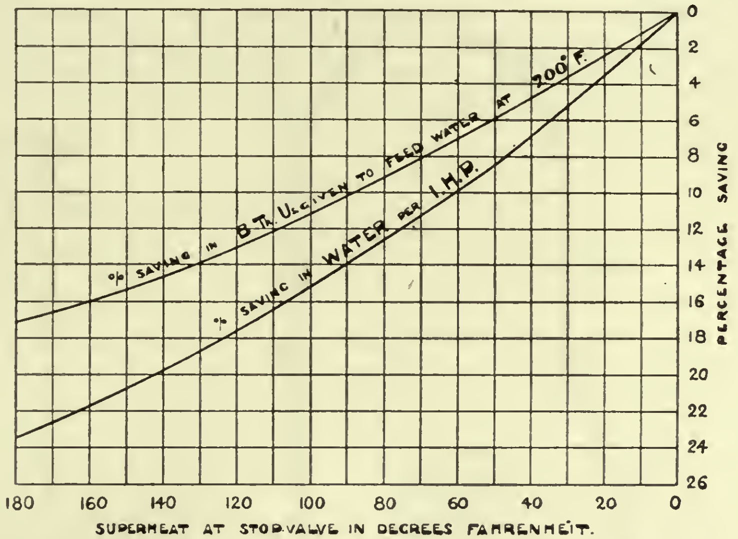


FIG. 234—Diagram showing Saving in Steam Consumption on Willans Compound Expansion Condensing Engine, with varying Degrees of Superheat.

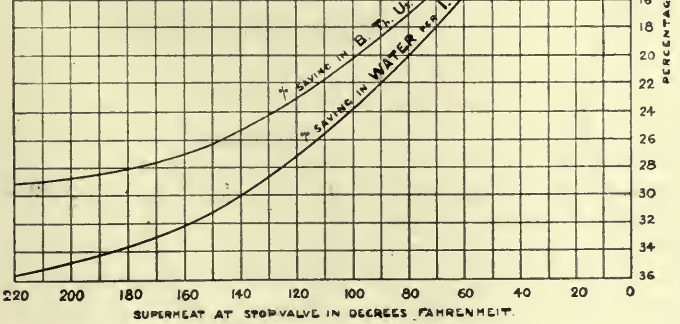


FIG. 235.—Diagram showing Saving in Steam Consumption on Willans' Simple Non-Condensing Engine, with varying Degrees of Superheat.

With the central valve engine, Messrs. Willans & Robinson have experienced no difficulty in working with the highest temperatures now in use. Many engines have been running in stations using steam at a temperature of 600° Fahr. with perfectly good results, and careful tests have been made up to temperatures of 700° Fahr. in order to ascertain their behaviour under any exceptional fluctuations of temperature. These have been equally satisfactory.

Due to the total absence of steam packing in the Willans engines, the special form of piston valves used, the total absence of metallic piston springs behind the rings, and finally to the very symmetrical form which all the parts of their engine assume, it is obvious that the Willans engine lends itself to the use of superheated steam more readily than other types.

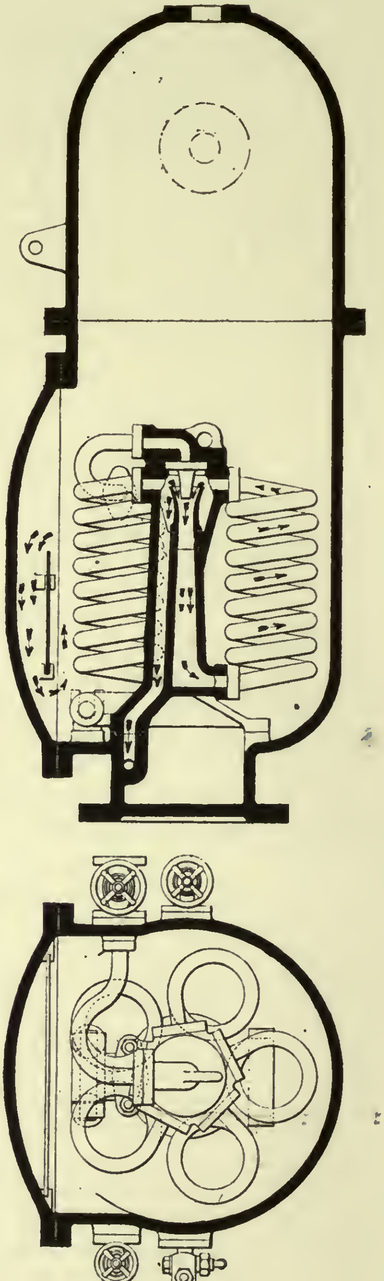


Fig. 236.

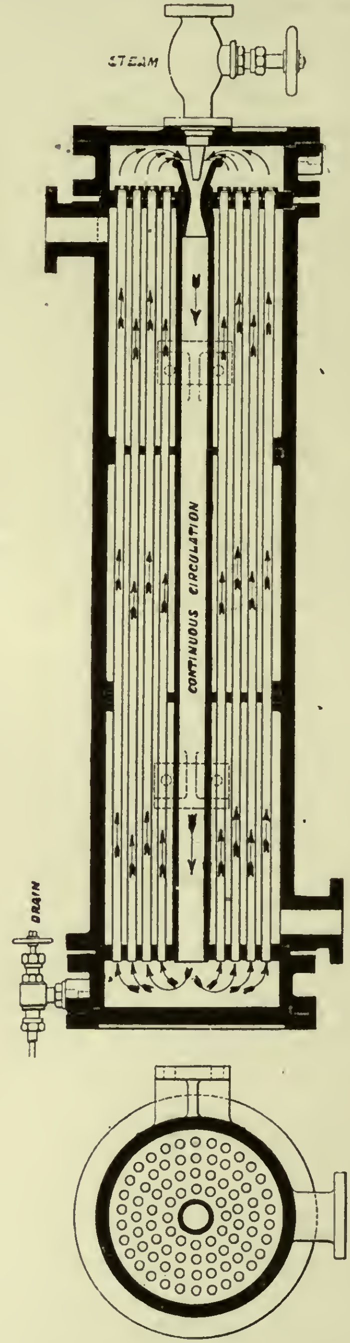


Fig. 237.

Andrews Steam Heater and Evaporator.

It might also be pointed out that the Willans engine will run equally satisfactorily with saturated steam or superheated steam, without alteration to the engine, and at the highest economy obtainable under the conditions prevailing. This is a matter of considerable importance, as it is found in practice that the temperature of the superheat is liable to vary—and in many engines it is found that the economy at the lower temperatures is sacrificed, due to the clearances having increased by a shrinkage in the valves, and in this way the chief advantage of using superheated steam is discounted. This falling off in economy is entirely avoided in the Willans engine, the leakage of steam past the valves being no greater at lower temperatures than at the higher ones.

ANDREWS STEAM HEATER AND EVAPORATOR

In this apparatus the principle of rapid circulation of the cold fluid to be heated and the hot fluid to carry the heat is made use of by introducing the injector principle.

The point aimed at by the designer, Mr. James Andrews, M.I.N.A., is the greatest possible mobility in the two fluids between which heat is to be interchanged. This mobility is, as many experiments have shown, more important in the case of the gaseous fluid than in that of the liquid one, since the rate of transfer of heat between a metal surface and a liquid in quiet contact with it is much greater than that between a metal and a gas or vapour under the same conditions. The construction of the apparatus will be readily understood on reference to Fig. 237. Steam is admitted at the top through a sort of injector nozzle as shown, and a rapid current of steam is thus established down the central tube and up the surrounding smaller tubes, as indicated by the arrows. The drain is placed at the bottom of the heater, and by opening this to a greater or lesser extent the flow of steam through the apparatus can be readily adjusted without touching the steam admission valve at the top. The scouring action of the steam current is stated to keep the tubes clean. Any condensed steam which may collect is also put in circulation, transferring its heat to the tubes, and taking up a fresh supply from the incoming steam. By means of the baffle plates shown the water outside the tubes is caused to follow a zigzag path across the tubes. It enters at the top and escapes below through the branch shown. When the original experiments were being made by Mr. Andrews it was considered that if water was substituted for steam inside the tubes the rate of heat transmission would be greatly increased. The heater was therefore turned upside down, so that the injector was at the bottom and the drain at the top. Steam was then turned on with the tubes full of water; the steam condensed in the water, heating it, and also putting it into a rapid motion. It was found that although the injector was not of the best design for circulating water, the rate of heat transmission was 39 per cent. better than when vapour alone was passing through the tubes. The steam pressure used in this experiment was 46 lbs. per square inch, and that at the drain pipe was atmospheric.

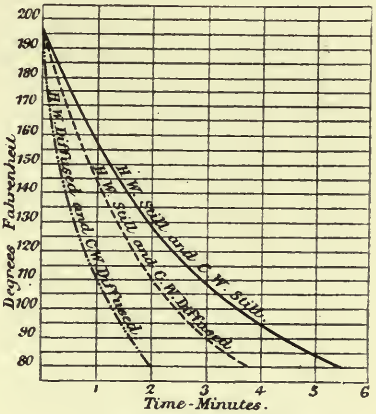


FIG. 238.—Diagram of Heat Transmission.

A sea-water evaporator embodying the same principle is shown in Fig. 236. Here the heating tubes are coils secured to a central discharge tube by simple flanges. Facilities are provided, as usual, for removing each coil separately, and as the weight of each is carried by the bottom of the evaporator the joints are free from undue strain.

In Fig. 238 are represented the results of an experiment of Mr. Andrews, showing the importance of securing the circulation of two fluids between which heat is to be interchanged. A tin of hot water was immersed in a basin of cold water, and the time taken for the temperature of the hot water to fall from 197° to 80° Fahr. under three conditions—namely, (*a*) whilst both the hot and cold water were left in quietude; (*b*) whilst the cold water was stirred or diffused; and (*c*) whilst both were stirred. The curves show that in the latter condition the rate of cooling was nearly three times as fast as it was when both liquids were quiet.

WATER SOFTENERS

Another mechanical and chemical process sometimes required in steam plants is a water purifier or softener. All waters contain more or less soluble salts of the earthy metals—lime, magnesia, etc., and these deposit on the inside of boilers, pipes, and other places as scale. And this scale is a poor heat conductor, and so retards the passage of the heat through the plates of the boiler or walls of the tubes. And the consequence is that the difference of temperature between the flame side of the plate and the water in the boiler is unduly great, and there is positive danger of weakening the plates. It also, by interfering with the passage of heat, creates unequal expansions and strains in the flues and plates, particularly over the furnaces.

Grease seems in some not very clearly defined combination to form a scale when it enters a boiler. It could not of itself form a scale, for at boiler temperatures oils are perfectly fluid, and would float. There is now no necessity to use animal fats or vegetable oils for lubrication,—these form corrosive compounds; but the heavy mineral oils now used seem to form an emulsion or soapy compound which is infusible at boiler temperatures, and sticks to the plates in thin scale or film.

The separation of grease by filtration through wood shavings, canvas cloth, and other devices is not successful; besides, these filters require continual cleansing.

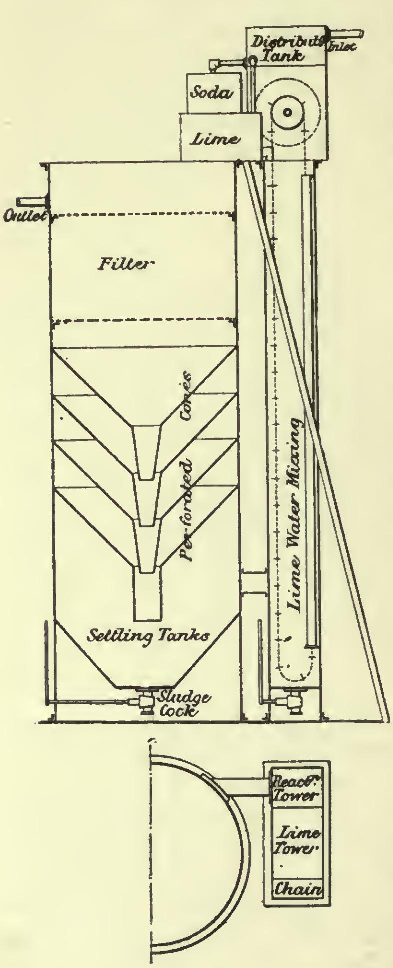


Fig. 239.—Stanhope Water Softener.

THE STANHOPE WATER SOFTENER

The author had some experience of one of this type on very foul water, with which it dealt successfully and at little cost. It is shown in Fig. 239.

The apparatus (Fig. 239) consists of a lime-mixing tank, with a small compartment for the mixing chain, a reaction tank, and a settling tank with inverted perforated funnels and a filter. Above these cylindrical tanks is a distributor, a soda tank, and a lime-slaking tank. The distributor has three carefully adjusted outlets, one discharging into the reaction tank, and one into the lime tank, and another into the soda tank. The chain in the lime-mixing tank is driven by a water wheel.

Capacity.—2500 gallons per hour.

Dimensions.—Settling tank, 7 feet 6 inches in diameter over angles; reaction and settling tower, 5 feet 6 inches by 2 feet 6 inches over angles; height, 26 feet; floor space, 70 square feet.

Working.—The water enters the distributor, the supply being regulated by a ball tap at the discharge. Three carefully gauged outlets lead this water respectively into the mixing tank, into the lime-mixing tank, and into the soda tank. The slaking of the lime is done in a separate tank. The reaction between the lime, soda, and untreated water takes place at the same time. The treated water passes upwards through perforations in the cones, and then through the filter. The sludge falls down through the openings in the centre of the cones, and is run out of the sludge cock.

Chemicals used per 1000 *Gallons.*—0.5 lb. lime, 0.5 soda crystals.

In a paper read before the Institute of Mechanical Engineers, dealing fully with the whole subject of water softening apparatus by Messrs. Stromeyer & Baron, the chemical aspect of the subject is fully treated. The following chemical actions and reactions are abstracts therefrom :—

Carbonate of Lime is the Chief Cause of Temporary Hardness.—Its chemical formula is $CaCO_3$. It is practically insoluble in water, but it is easily converted into bicarbonate of lime, having the formula $CaH_2(CO_3)_2$, which is fairly soluble in cold water, and is a constituent of most natural waters. Its second equivalent of carbonic acid is easily removed on boiling, when, of course, the carbonate of lime is precipitated, forming a scale. The slower this reaction is carried out, the slower the heating, the more chance is there for this carbonate of lime to form crystals—called calcite in mineralogy, which constitute a fairly hard scale, as found in economisers, where the conditions are very favourable to its production. This explains why, in the south of England, where the waters contain much temporary hardness, economisers are very little used, for they would get choked unless the waters be first softened. Waters poor in carbonate of lime, but rich in sulphates, do not choke the economiser pipes. If the heating is effected rapidly, as, for instance, when natural waters are pumped direct into the steam space of a boiler, the precipitation of carbonate of lime is so rapid that only a mud is formed. Another means of converting bicarbonate of lime into carbonate of lime is to add solutions of caustic lime, burnt lime, or caustic soda in water. The chemical reaction is as follows :—

$$\begin{array}{ccccccc} \left.\begin{matrix}\text{Bicarbonate}\\ \text{of calcium}\end{matrix}\right\} & + & \left\{\begin{matrix}\text{Caustic}\\ \text{lime}\end{matrix}\right\} & = & \left\{\begin{matrix}\text{Carbonate}\\ \text{of lime}\end{matrix}\right\} & + & \text{Water} \\ CaH_2(CO_3)_2 & + & CaO & = & 2\,CaCO_3 & + & H_2O \\ \text{Soluble} & & \text{Soluble} & & \text{Insoluble} & & \text{Water} \end{array}$$

The second equivalent of the carbonic acid in the bicarbonates of lime and of magnesia is generally called half-bound. Any excess is called free carbonic acid. Of course, sufficient lime is required to neutralise both.

In the table of chemical analysis the bicarbonate of lime is split up into carbonate of lime and carbonate acid, because the latter disappears on boiling. One grain of carbonic acid combines with 2.27 grains of carbonate of lime, forming 3.27 grains of bicarbonate of lime. Temporary hardness cannot be entirely removed by lime treatment, because carbonate of lime is not absolutely insoluble.

Sulphate of Lime.—The chemical formula is $CaSO_4$. It is fairly soluble in waters up to the boiling-point at atmospheric pressure, but it is less soluble at temperatures corresponding to high pressures of steam. Because this sulphate of lime cannot be removed by ordinary boiling, it is called permanent hardness. One grain of sulphate of lime equals 0.7350 permanent hardness. The result is that if waters containing sulphate of lime are pumped into a boiler, the constant evaporation effects a slow concentration until saturation-point is reached, when the sulphate of lime is precipitated so slowly that it crystallises and adheres to all parts of the boiler, but more particularly to the hottest parts ; then, whenever the pressure drops—and with it the temperature—part of the precipitate is redissolved, the space between the crystals are filled with concentrated solutions of sulphate of lime, which crystallises out again on heating, whereby the originally small and loose crystals are enlarged and are firmly cemented together, forming a very hard scale, which can only be removed by chipping or by heating ; it is practically gypsum or selenite. If any carbonate of lime is mixed up with the scale, this too gets thoroughly cemented with it.

At the temperatures corresponding to high pressures, water can dissolve only about 20 grains of sulphate of lime per gallon. On cooling down a boiler and letting it stand for some time the water will dissolve parts of the crystals until there are about 170 grains to the gallon. This dissolving action loosens the scale, which can be easily removed as long as it is wet. If the scale be allowed to dry all the 170 grains of sulphate crystallise, and thereby cement together the loose parts of the scale, making it hard and difficult to remove.

The principle of pumping feed water into a trough in the steam space of a boiler has several times been patented, but as boiler water can hold up to 20 grains of sulphate of lime in solution this method is of no benefit with waters having less permanent hardness than 15°.

The sulphate of lime can always be entirely removed by conversion into carbonate of lime. The reaction is as follows:—

$$\begin{array}{ccccccc} \left.\begin{matrix}\text{Sulphate}\\ \text{of lime}\end{matrix}\right\} & + & \left\{\begin{matrix}\text{Carbonate}\\ \text{of soda}\end{matrix}\right\} & = & \left\{\begin{matrix}\text{Carbonate}\\ \text{of lime}\end{matrix}\right\} & + & \left\{\begin{matrix}\text{Sulphate}\\ \text{of soda}\end{matrix}\right. \\ CaSO_4 & + & Na_2CO_3 & = & CaCO_3 & + & Na_2SO_4 \\ \text{Soluble} & & \text{Soluble} & & \text{Insoluble} & & \text{Soluble} \end{array}$$

If there is any free or half-bound carbonic acid in the water—and this is generally the case—caustic soda, if available as a waste product, may be used. The caustic soda combines with the carbonic acid to form carbonate of soda, and the reaction is then as sketched above; and the lime which has lost its one equivalent of half-bound carbonic acid is also precipitated. Caustic soda without carbonic acid will not precipitate the sulphate of lime. It is therefore wrong to introduce caustic soda into boilers which are fed with water having only permanent hardness. In fact, caustic soda should not be pumped into a boiler except together with the cold feed; for as long as the water is cold the carbonic acid can be fixed, and additions of caustic soda would come too late if made in the boiler.

The above sketched chemical reaction can easily be carried out in water softeners, but the separation of the precipitate is as slow as in the last case; and, unless heat is applied, the newly formed carbonate of lime has to settle for a long time.

Magnesium Salts.—These are found in natural waters as nitrates, chlorides, sulphates, and cause permanent hardness, or as bicarbonate, which causes temporary hardness; they are very soluble, except the last-named. They all react on soap, and heat will not precipitate them, except the bicarbonate, by driving off its carbonic acid. At the high temperatures to be found in boilers a reaction resulting in precipitation of magnesia takes place between the carbonate of lime and the soluble magnesium salts. The bicarbonate of magnesia can be partly removed by converting it into carbonate of magnesia by the addition of caustic lime.

The reaction in water softeners is as follows:—

$$\begin{array}{ccccccccc} \left.\begin{matrix}\text{Bicarbonate of}\\ \text{magnesia}\end{matrix}\right\} & + & \left\{\begin{matrix}\text{Caustic}\\ \text{lime}\end{matrix}\right\} & = & \left\{\begin{matrix}\text{Carbonate of}\\ \text{magnesia}\end{matrix}\right\} & + & \left\{\begin{matrix}\text{Carbonate}\\ \text{of lime}\end{matrix}\right\} & + & \text{Water} \\ MgH_2(CO_3)_2 & + & CaO & = & MgCO_3 & + & CaCO_3 & + & H_2O \\ \text{Soluble} & & \text{Soluble} & & \text{Slightly soluble} & & \text{Insoluble} & & \text{Water} \end{array}$$

The last traces of magnesium carbonate are removed by adding an excess of lime, the reaction being as follows:—

$$\begin{array}{ccccccccc} \left.\begin{matrix}\text{Carbonate of}\\ \text{magnesia}\end{matrix}\right\} & + & \left\{\begin{matrix}\text{Caustic}\\ \text{lime}\end{matrix}\right\} & + & \text{Water} & = & \left\{\begin{matrix}\text{Hydrate of}\\ \text{magnesia}\end{matrix}\right\} & + & \left\{\begin{matrix}\text{Carbonate}\\ \text{of lime}\end{matrix}\right. \\ MgCO_3 & + & CaO & + & H_2O & = & Mg(OH)_2 & + & CaCO_2 \\ \text{Slightly soluble} & & \text{Soluble} & & \text{Water} & & \text{Insoluble} & & \text{Insoluble} \end{array}$$

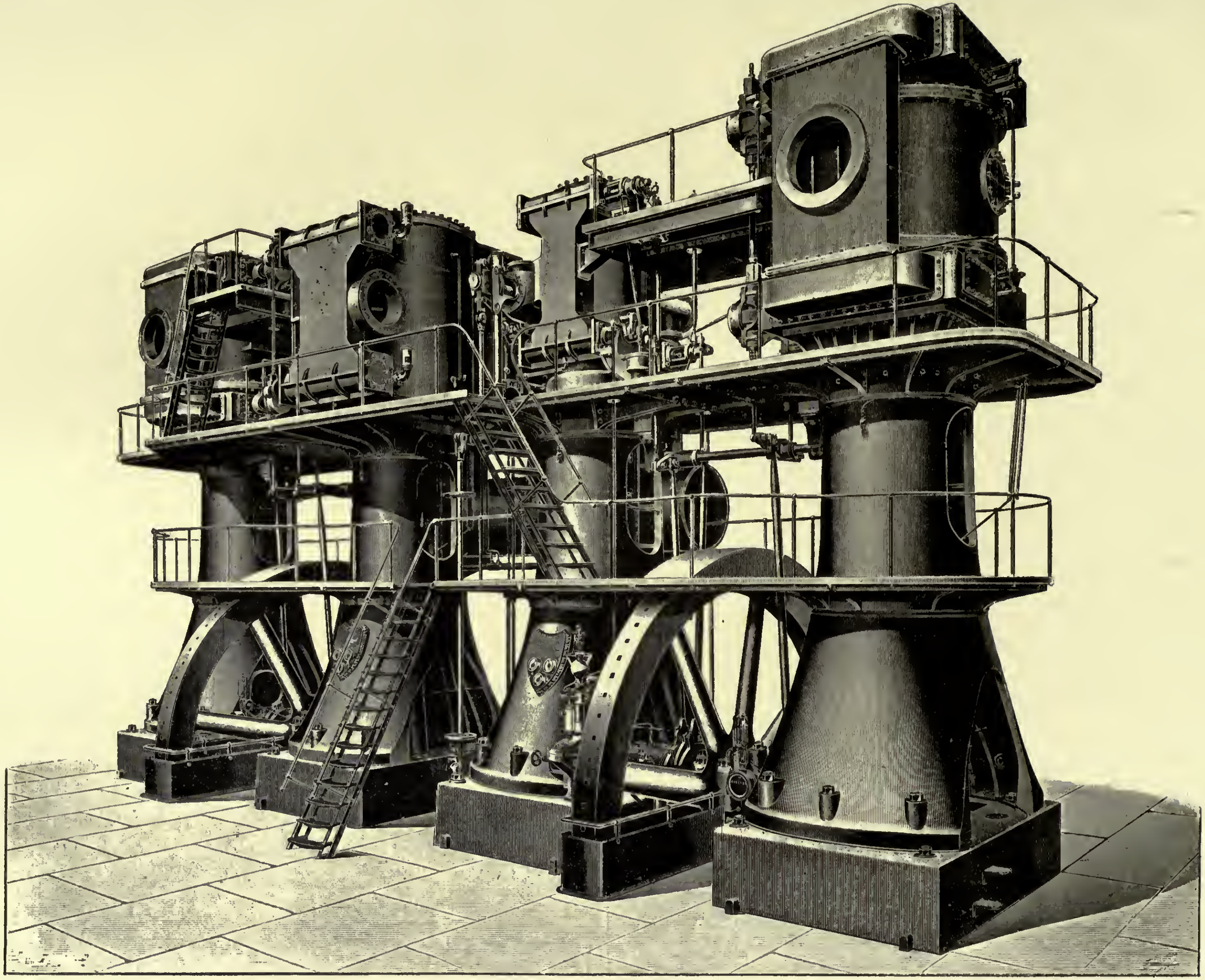


Plate V.—Quarter Crank Compound Blowing Engine.

The other magnesium salts react in a similar way, being easily converted into carbonic and hydrate by the addition of burnt lime and soda ash.

The carbonate of magnesia ($MgCO_3$) is very slightly soluble in water, whereas the hydrate of magnesia ($Mg(HO)_2$) is practically insoluble ; but being a gelatinous mass, it causes serious difficulties in water-softeners, more especially by clogging the filters.

As an example of a water softening apparatus, a common type is shown in Fig. 240.

The apparatus (Figs. 240–241) consists of a chemical tank, a reaction tank, filters, and soft-water tank. The supply pipe passes down the side of the chemical tank, and ends in the bottom of the reaction tank. It has a branch pipe and cock over the chemical tank, and is fitted with valve V, shown on larger scale, which is regulated by the float in the soft-water tank. Live steam enters the reaction tank as shown, through a silent injector ; a branch from the steam pipe for occasional use is led into the bottom of the chemical tank. Where exhaust steam is available an additional injector is fitted as shown. Attached to the bottom of the chemical tank is a pipe with two cocks—C_1 for regulating the ratio of soda solution to the feed, and C_2 for regulating the quantity in accordance with the feed drawn off. At the top of the reaction tank is a weir, over which the treated water flows into the filter, which consists of a series of boxes filled with wood wool. The bottom of these boxes are provided with cocks, through which the sludge is drawn off occasionally. The filtered water passes in the soft-water tank, whence it is pumped into the boiler.

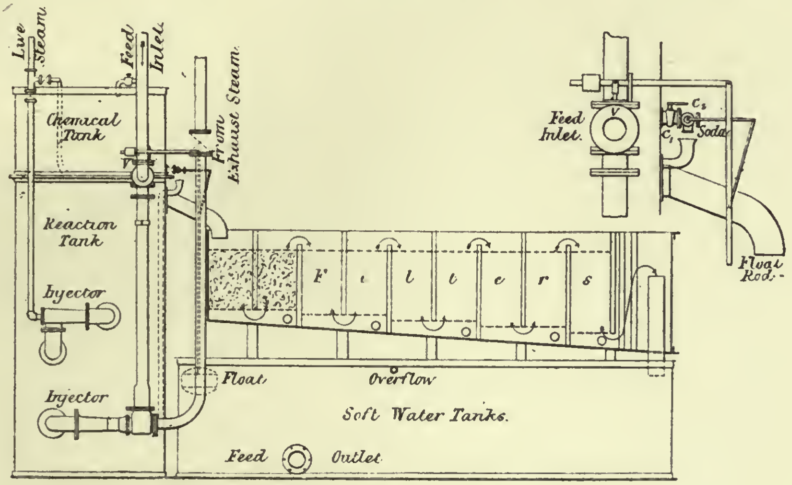


FIG. 240.—Babcock & Wilcox Water Softener.

Capacity.—5000 gallons per hour.

Dimensions.—Chemical and reaction tanks, 5 feet 5 inches square and 13 feet high. Filters and soft-water tank, 5 feet 5 inches wide, 17 feet long, and 8 feet high, measured over flanges. Total floor space, 122 square feet.

Working.—Enough soda to last for one day is placed in the chemical tank, water is added, and steam turned on till all the soda is dissolved. The cock C_1 is opened to suit the density of the soda solution and the impurity of the feed water, and has to be adjusted during working according to whether the softened water is alkaline or not. The cock C_2 and the feed valve V are actuated by levers from one float in the settling tank, so that the ratio of the amount of chemical and of water is maintained. The mixture flows into the reaction chamber, where it is mixed with waste steam if available. It is here brought to the boiling-point by live steam, which should remove the temporary hardness ; the soda removes the permanent hardness. The hot water overflows to the filters, which it passes in half an hour, and here the sediment is removed. Some sediment collects in the lower tank. The grease from the exhaust steam adheres to the sediment, and does not pass the filters.

This water softener requires no attention or labour, except to fill the chemical tank every morning. It works quite automatically, a float regulating the admission of hard water and soda solution in proportion to the quantity of soft water consumed. It can be used for any process of water softening, whether lime, caustic soda, or other chemicals are employed, but the use of carbonate of soda (alkali) is preferable. It will remove all carbonates and sulphates, whether of lime or magnesia, in a thoroughly efficient manner, and the softened water will not attack the boiler plates or fittings, as is the case with many other processes.

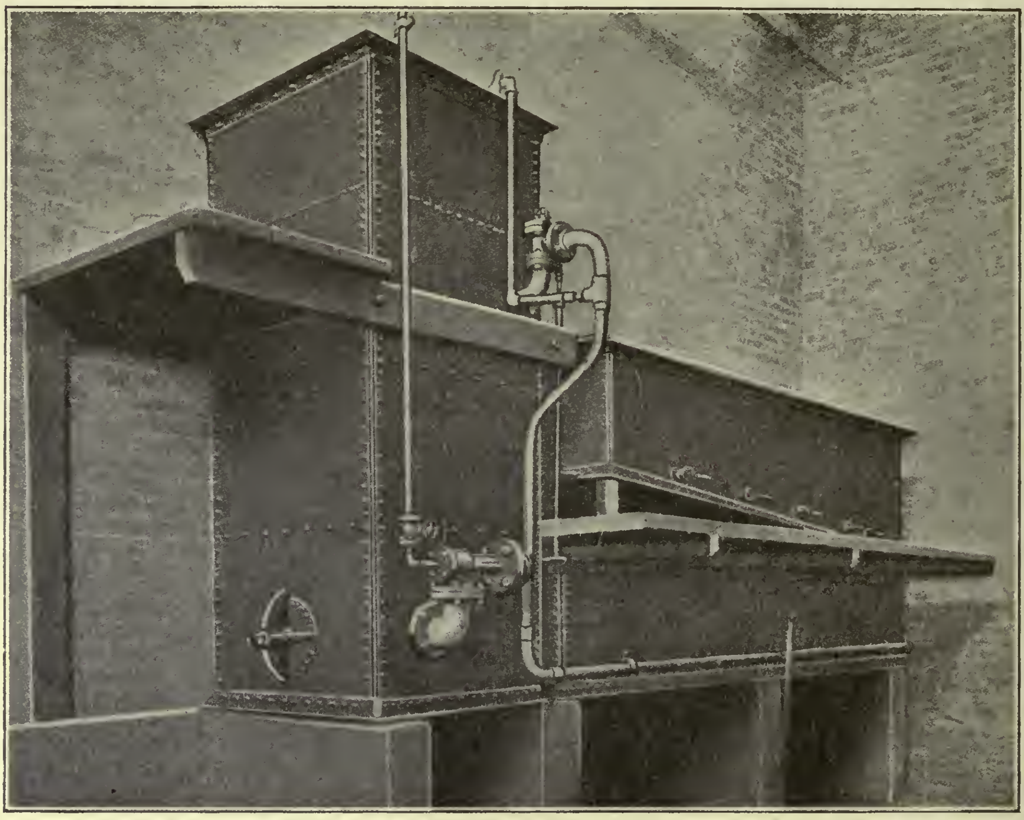


Fig. 241.—Babcock & Wilcox Water Softener.

The Magdeburg Steam Boiler Society, which is one of the largest of its kind, states in a recent circular: "No patent remedy against boiler scale is more effective than soda, and all are much dearer."

The following examples show the result obtained with East London water treated in this apparatus for two Babcock & Wilcox boilers:—

	Hard Water, grains per gallon.	Softened Water, grains per gallon.
Temporary hardness . . .	14.2	3·7
Permanent	3.1	None
Total	17·3	3·7

Of the 3.7 grains per gallon, 2.7 grains are further precipitated by prolonged boiling in the boiler itself; the remaining 1.0 grain represents the solubility of calcium and magnesium carbonates.

During an experience extending over three years no scale whatever was discovered in the boilers. After working a boiler continuously without cleaning for ten months the only scale found was in the lower row of tubes, these having a thin layer of loose deposit, which by a jet of water from a hose pipe was easily removed.

The consumption of alkali was 2 ounces per 1,000 gallons of water for each degree of hardness, and the cost of softening 1000 gallons of East London water was found to be only one penny.

The apparatus requires cleaning out about once a month only, and this can be done whilst it is working. The filtering material is used again, and does not require renewal for years. Any quantity of water from the smallest to the largest can be treated equally as efficiently.

The only effective way of dealing with grease is to collect or blow off the bulk of it, which collects and floats on the water, and to deal chemically with the emulsified portion of oil remaining in the water.

A large portion of the oil in the feed water may be separated in a properly constructed feed-water heater. A heater designed with this in view is shown in Fig. 242, a section of Morison's feed-water heater. The heating is accomplished by coils and boiler steam. Both air and oil are separated from the water in this apparatus, and they collect in the dome of the apparatus. The air is discharged automatically by an air valve, shown at the right-hand side. When it is necessary to clean or examine the float valve the two cocks connecting it with the heater should be shut off, and on restarting the float casing should be recharged with air from the dome reservoir by first opening the top cock.

If feed water containing oil is heated the oil floats on the surface in the form of scum. The feed water on its upward course through the heater is rapidly raised in temperature, and on passing through the contracted area at the edge of the inverted cone its velocity is so increased that the separated oil flows to the surface, where it accumulates and remains, its buoyancy and surface tension overcoming the tendency to be drawn down the inverted cone. Once each watch the cock on the top of the dome reservoir is opened, and the air above the normal water level being released, causes the floating oil to be gradually concentrated by the diminishing area of the long cone until it is finally discharged into a tank with a minimum loss of feed water.

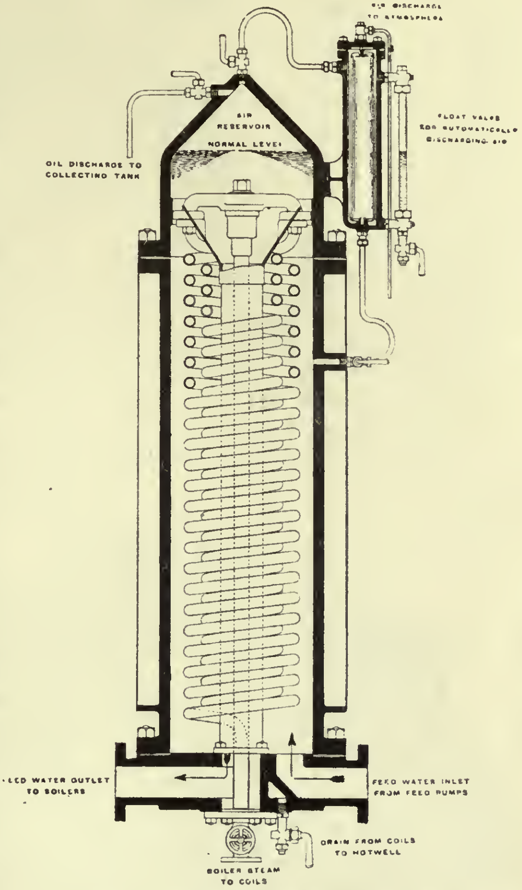

AIR DISCHARGE TO ATMOSPHERE
AIR RESERVOIR
NORMAL LEVEL
FLOAT VALVE FOR AUTOMATICALLY DISCHARGING AIR
OIL DISCHARGE TO COLLECTING TANK
FEED WATER OUTLET TO BOILERS
FEED WATER INLET FROM FEED PUMPS
DRAIN FROM COILS TO HOTWELL
BOILER STEAM TO COILS

FIG 242.—Feed Heater and Oil Separator.

Before discharging the oil the float casing should be shut off by means of the two cocks, and these cocks should not be reopened until air has accumulated in the dome. When the float casing is drained it should be put in connection with the heater by first opening the top cock, as, the float being open-bottomed, air is necessary to re-establish buoyancy.

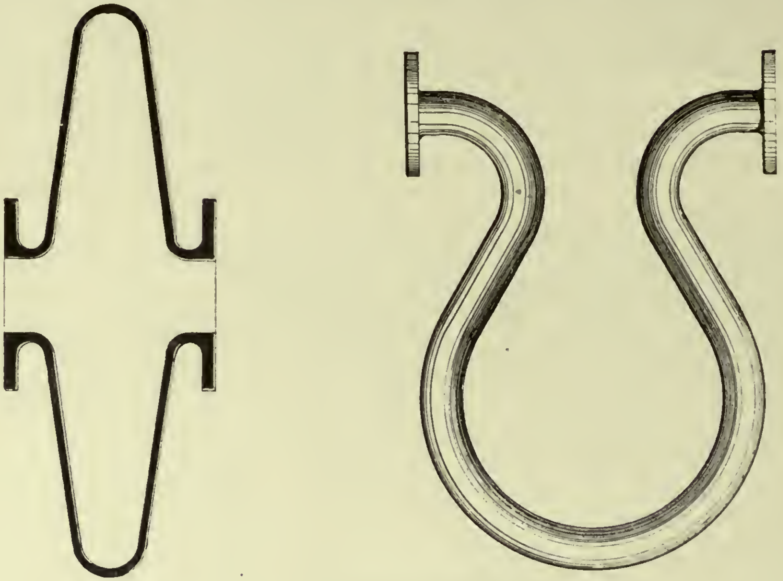


FIG. 243.—Copper Expansion Bends.

The heaters are manufactured in various sizes, and can be fitted with bypass valves if so required.

Before concluding this survey of the various parts and accessories of steam power

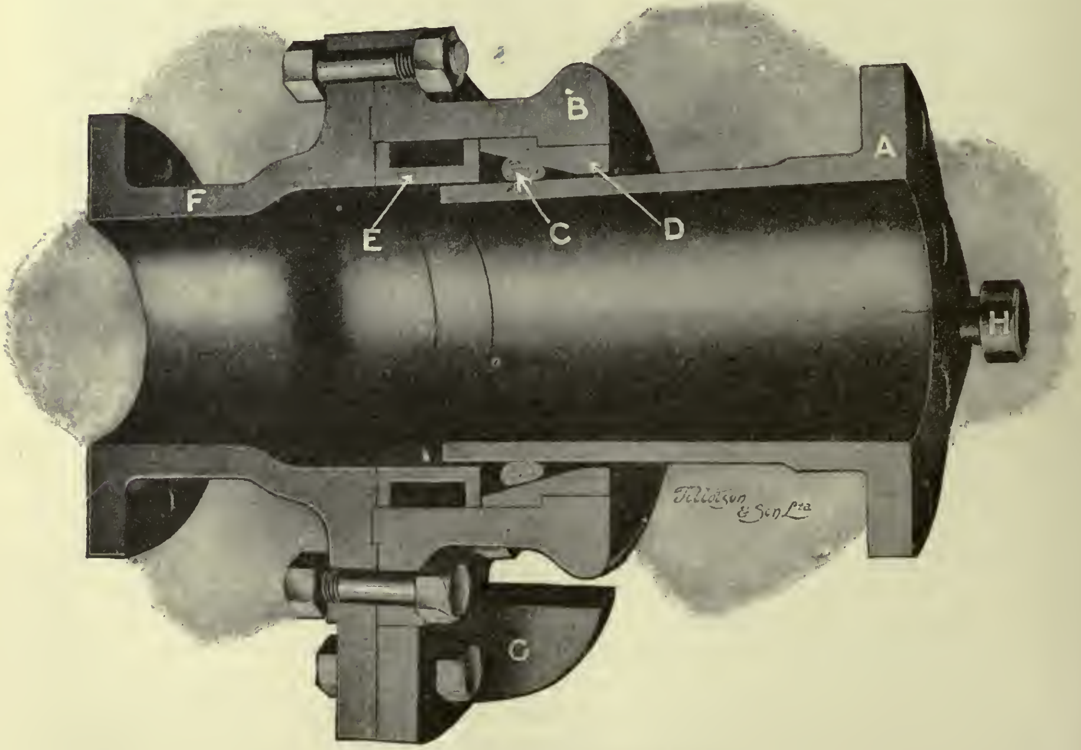


FIG. 244.—Geddes Expansion Joint.

plants it is necessary to include flexible steam joints as necessary devices for safety in high-pressure steam pipes; and also flexible pipes and expansion glands for the same purpose. A bent copper pipe gives sufficient elasticity for most land engines, and it is usual to make this pipe the closing piece near the engine. And similar bends may be inserted in the pipes wherever expansion and contraction has to be allowed for. This simple bend is shown in Fig. 243. Another copper flexible joint (Fig. 243) consists of a flat vessel which expands or collapses like a bellows. These, however, do not allow of much movement. In cases where the motion is considerable, flexible or expansion joints are necessary.

GEDDES EXPANSION JOINT

The construction, which will be easily understood from the illustration (Fig. 244), is as follows:—To one end of the steam pipe is bolted a short tube, which terminates within a conical gland containing an endless packing ring. The steam pressure acting on this ring forces it into the conical space, thus preventing the escape of steam. When it is required to examine or renew the packing ring the bolts of the centre flange are disconnected, and the gland drawn towards the left hand over the tube. The loose ring, which is in segments, and which serves to keep the packing ring in place, can be then removed, and the packing ring drawn towards the right hand over the end of the tube. The outer end of the gland being a free fit for the pipe allows for universal movement, whilst the friction between the packing and the pipe is so reduced that there is perfect freedom for lateral movement.

THE OLIVER EXPANSION JOINT

Figs. 245 and 246 illustrate an expansion joint, the principal feature of which is the bearing which guides the expansion sleeve, taking all the strain off the stuffing box, which prevents leaking. The fixed end serves as an anchor for the pipe, dispensing with the clamps commonly used. The design is such that it is impossible for the expansion sleeve to pull out, and as the interior of the bearing is babbitted, the friction is materially reduced and rusting prevented. It is manufactured by the C. G. Bretting Manufacturing Company, of Ashland, U.S.A.

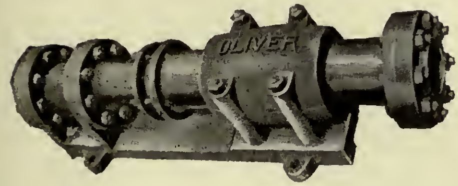


Fig. 245.

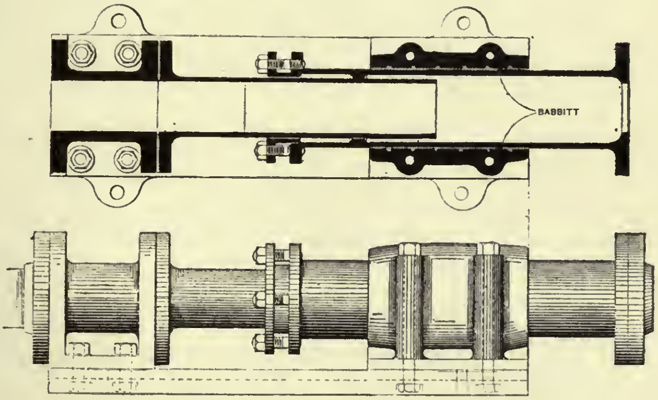


Fig. 246.

Oliver Expansion Joint.

HARTER'S FLEXIBLE UNIVERSAL STEAM JOINT

This well-designed device is in use in the Navy in several vessels. It is made in two types, as a universal joint and as an expansion joint. It will be noticed that, unlike most other kinds of ball joint, the ball pieces A are held in place by outside stays, which are pivoted to the socket pieces B, and which resist any tendency of the ball pieces to blow out of the sockets, no matter how high the pressure, while at the same time the joint is free to swing with very little friction. Steam-tightness is provided for by an ordinary gland and stuffing box as shown, which is entirely independent of the means of holding the ball in place.

Referring to the universal joint type (Fig. 248), it will be seen that the stays are pivoted on to a steel ring at CC, while this ring is itself pivoted on to the socket piece at DD. The pivots DD give the ring, stays, and ball piece freedom to move in a plane parallel to the paper in Fig. 248, while the pivots CC give the stays and ball piece freedom to move in a plane at right angles to this. The axes of the pivots, of course, intersect at the centre of the ball; hence the ball piece is free to swing in any direction and with a minimum of friction, however high the steam pressure.

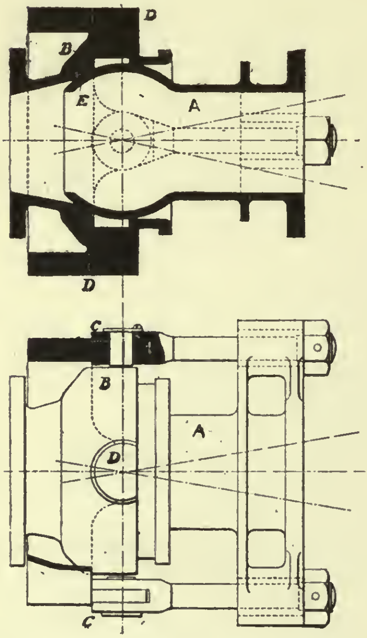


FIG. 247.—Harter's Expansion Joint.

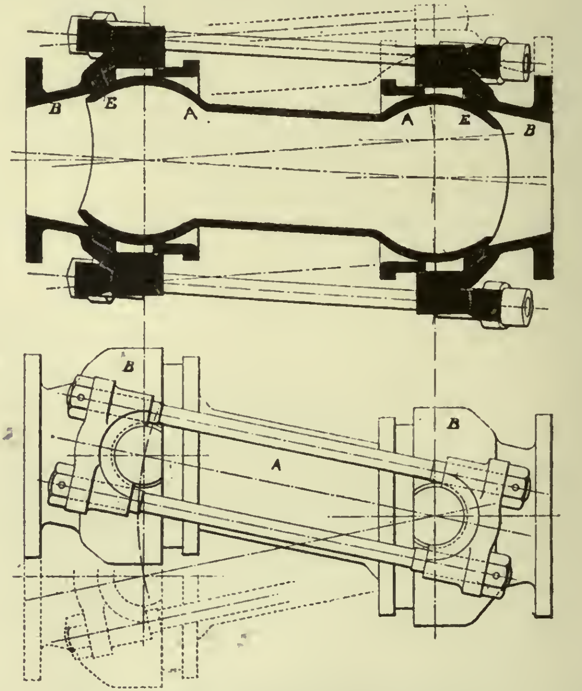


FIG. 248.—Harter's Universal Joint.

In the expansion joint type (Fig. 247) the steel ring is dispensed with, and the stays, which for convenience are made double, are pivoted direct on to lugs on the socket pieces as shown. These lugs are made spherical, in order to allow of slight movement, as shown in Fig. 247, the axis of the lugs in one socket piece remaining parallel to the axis of the lugs in the other. The principal movement is, however, in a plane at right angles to this.

In Fig. 249 it will be seen that while one socket piece is bolted direct to the stop valve on the boiler, the other is bolted to the main pipe. As the pipe expands the end moves in the direction of the arrow, and the pair of ball joints at each boiler act as swinging links connecting the boilers to the main pipe in a perfectly flexible manner.

This movement is the same as shown enlarged in Fig. 248, while the movement that takes place in Fig. 248 is that due to the top of the boiler itself rising with its own expansion. It will be seen that these joints can be arranged to give a connection to a boiler, or engine, etc., that is absolutely free in any direction, as well as taking up the expansion of the pipes, without causing any unbalanced end thrust whatever. The piping is all in one plane, there being no vertical bends, so that there are no pockets where water can collect. The horizontal bends may be made as easy as desired.

Water grooves are arranged on the inside of the socket piece, or in a separate bush, as shown at E. This is always in contact with the ball piece, and assists in securing steam tightness. There is also the stuffing box for soft packing. Owing to the fact that the width of the packing case can never change, it is a simple matter to keep the gland tight. This is not so with most stuffing boxes, as, although they may be quite tight as long as perfect alignment is maintained, they cannot be kept so where there is any side-play. There must, of course, always be perfect alignment with the ball joint, on account of the working surface being part of a true sphere. It is frequently the case that the principal strains on piping and its joints are not due to the internal pressure as much as to a bending action on the pipes caused by expansion and contraction, or separate movement of the parts connected, due to vibration, and the like.

The many accessories to the steam plant when it is to be run economically have now been described. It is when the steam engine, with all these attendant auxiliary assistants, is compared with a gas engine and gas producer that a fair comparison may be made between the two plants in respect to complicated mechanism and multiplicity of parts.

In steam plants the engine itself is a small part of the whole, and its efficiency does not greatly influence the value of the total efficiency of the plant. Nevertheless, the ordinary steam user, in laying down a steam plant, nearly always considers the engine only, and takes all the rest on trust.

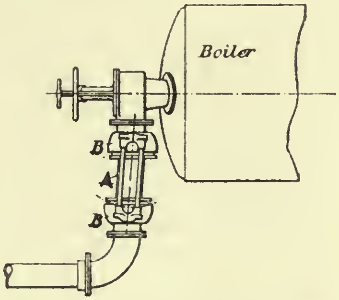


FIG. 249.—Ball and Socket Joint.

We will refer to the difference between the total efficiencies of steam and gas plants fully in the volume on Power Transmission. Meanwhile the consideration of modern forms of steam engines may now be taken up.

CHAPTER IV

MODERN TYPES OF STEAM ENGINES

Nowhere in all the range of human experience is better illustrated the natural inertia of mankind to move beyond old boundaries, or what is called good practice, than in the history of the steam engine.

It is the same in all great human advances through science and invention, but in steam engineering it is more conspicuous. Steam having been *the* prime mover of the eighteenth century, the results of the establishment of the steam engine by Boulton and Watt have been tremendous ; but its progress from the early types to the modern engine was slow. Even Watt himself was so satisfied with the stage at which he arrived that he positively denounced the use of higher pressures and other aids to economy.

The British engineers enjoyed the advantages of isolation from excursions and alarms of war during the whole century of the steam engine's progress, and therefore had nothing to interfere with the manufacture for home and abroad. However, up to the conclusion of the great struggle in the Franco-German War, which paralysed the Continent of Europe for the time, and for some time afterwards, the British engineer had no competitor of any consequence, and fortunes were made producing engines, without going to the trouble and expense of improvements in principles or details. The attractive problems were mostly regarding the adaptation of the engine to different purposes. There is, as a fact, more fame and fortune won by the adaptor of well-known things to new applications than by the original inventors. And thus we find several names celebrated for different applications of steam power. One puts a boiler and engine on a boat working paddles, and becomes famous as the inventor of the steamboat. Another puts the boiler and engine on four wheels, and becomes famous as the inventor of the locomotive.

Watt's engine dates back a hundred and fifty years or so, yet little progress was made for the first hundred years. The progress thereafter made was more due to improved machine tools and better material, enabling higher pressures to be used. After the mid-century wars were over, France, Germany, and America settled down to peaceful industry, and commenced to compete with us, instead of purchasing from us, and that competition is as keen now as it is between home manufacturers themselves. This competition was the spur to improvements during the past thirty years, which has brought the reciprocating steam engine of modern design and construction to such a state of completion and efficiency that there is absolutely no prospects of further improvement. But just at the moment of its highest development the steam turbine appears on the scene, and bids fair to oust the reciprocating engine at a very early date.

Here again there is strong evidence of inertia, among those who should be easily moved to abandon old and adopt new inventions. All the essential features and details of any known steam turbine are free and unprotected, the patents expired in many cases, even before the invention was heard of. I myself saw Parsons' turbines successfully at work in 1888, and De Laval's in 1894. Yet only since 1900 have they been generally known to engineers. And even now it is looked upon as a bit of adventure to adopt a steam turbine as a prime mover. In this country the number of steam turbines made and sold in 1904 is not a large figure, nor are any great preparations being made for their production; but the two great Anglo-American firms, the British Westinghouse Company and the British Thomson-Houston Company, are vigorously pushing the steam turbine business. And great developments are going forward in France and Germany, so that in a few years time the reciprocating steam engine makers are in for a keenly contested competition, which can end only in the triumph of the turbine.

On the question of high versus low-speed engines for factory or mill driving it may be observed that slow speed is the common practice, more because it has been the practice in the past than that there is any proof of its superiority. Two engines may have the same piston speed, and yet the one may revolve at many times the revolutions per minute of the other.

The slow revolution engine is gradually disappearing except for special purposes, such as heavy water pumping, in which case the speed has to be kept down to that which will operate the large volumes of water without shock and great losses.

The water, as a rule, has to be started from a state of rest, accelerated, and passed through valves, and lifted to a position of rest again. The motions must be slow, gradual, and steady; hence an engine of slow revolutions, large and massive, must be adopted. Our Frontispiece to this Volume shows an example of this class of engine in the large pumping engines there depicted.

This illustration is from a photo from engines in the erecting shop, and shows two engines of approximately the same horse-power, both built by Yarrow & Co. of Poplar. The comparison conveys a striking idea of the role which speed and pressure play in the generation of power. The large machine is a pumping engine built for the East London Water Works Company, and restricted, of course, in the speed at which it may be run by the conditions of pumping service. It is a triple-expansion engine, with cylinders 20, 32, and 53 inches in diameter and a stroke of 42 inches, running at 16 revolutions per minute with 150 lbs. pressure. The small engine was built for torpedo-boat service. It is also a triple-expansion with cylinders 8, 12, and 17 inches in diameter and a stroke of 9 inches. It runs at 550 revolutions per minute, with a pressure of 250 lbs.

The weight of the large engine is about 80 tons, and of the smaller about 2 tons. The rated capacity in both cases is about 250 horse-power.

The main features of both engines are compared in the following tables:—

	Pumping Engine.	Torpedo Boat Engine.
Diameter high-pressure cylinder . .	20 inches.	8 inches.
,, intermediate cylinder . .	32 inches.	12 inches.
,, low-pressure cylinder . .	53 inches.	17 inches.
Stroke	42 inches.	9 inches.
Revolutions per minute	16	550
Piston speed	112 feet.	825 feet.
Displacement of low-pressure cylinder per minute	1716 cubic feet.	1300 cubic feet.
Displacement per H.P.	6.86 cubic feet.	5.2 cubic feet.
M.E.P. required for 250 H.P. . .	33.4 lbs.	44 lbs.
Total weight	80 tons.	2 tons.
Rated capacity	250 H.P.	250 H.P.

The mean effective pressure in the low-pressure cylinder required to develop 250 horse-power is rather high for a triple-expansion pumping engine, even with 150 lbs.

initial pressure. As the mean pressure is proportional to the initial with the same ratio of expansion, the torpedo-boat engine with its 250 lbs. initial can easily realise the greater mean effective pressure required to develop the same power as the large machine.

Plate III. shows an elevation of another slow-speed water pumping engine by Messrs. Glenfield & Kennedy. The feature of interest in this engine is the method of direct transmission of the thrust of the steam from the steam piston to the pump plungers. They are both in line and connected to the one cross-head.

The connecting rods are long forks to clear the pumps between them. In this way the pressure is communicated to the water direct with transmission through the cranks and bearings. This principle has been applied to many blowing and pumping engines.

The modern pumping engine is, however, assuming the form of turbines, or centrifugal pumps. These are now made of high efficiency to lift any reasonable height, and as the water can flow smoothly, and no valves except a foot valve is necessary, they offer many advantages.

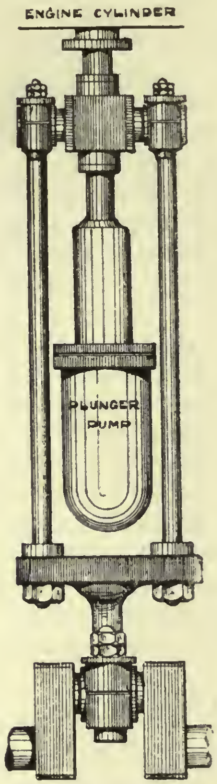


Fig. 250—Pump Plungers and Connecting Rod.

The ideas regarding centrifugal pumps which long prevailed assigned to them a very low efficiency, about 40 or 50 per cent., and little or no reason was given for the great loss, the general conviction being that this simple machine was of necessity wasteful of power. Modern engineers do not rest under such notions. When a great loss is manifest they want to know where, how, and why it occurs; then, if it is unavoidable and cannot be cured, it must of course be endured, but in simple mechanical operations great losses can in most cases be reduced to small losses by thoroughly mastering the principles of action, discovering the cause of the loss, and then, by designing and if necessary experimenting, getting rid of the cause of the losses producing a more efficient and better machine. All this procedure, however, costs time and money, and hence, so long as the machines can be sold as they are, improvements are not made even when made quite obviously possible.

The centrifugal pump for half a century nearly was considered quite perfect in practice, with an efficiency ranging from 40 per cent. in small powers to 70 per cent. in large powers, and the 70 per cent. was obtained only under very special conditions of speed and head. And it is very doubtful if it ever was really obtained in practice. Another notion held regarding centrifugal pumps was that only about 20 feet lifts could be efficiently worked.

But all these old prejudices are disappearing. If a centrifugal is as well designed and constructed as a first class turbine, its efficiency at all lifts from 10 to 1000 feet is higher than that of plunger pumps. In larger sizes 85 per cent. can easily be obtained.

Coupled to a good compound or triple-expansion engine, and with a jet condenser in which some of the pump water does the condensing, it forms the best modern pump.

For sewage pumping and dredging, of course, the centrifugal pump is the only one applicable; except the screw pumping engine, which has been successfully applied in America.

Fig. 252 illustrates a modern pumping engine by Messrs. Alley & Maclellan, Sentinel Works. It consists of their enclosed high-speed vertical engine coupled directly to their centrifugal pump.

The increasing favour in which centrifugal pumping plants are now held is shown by the adoption of a large plant made by Messrs. Tangye, Birmingham, for the East

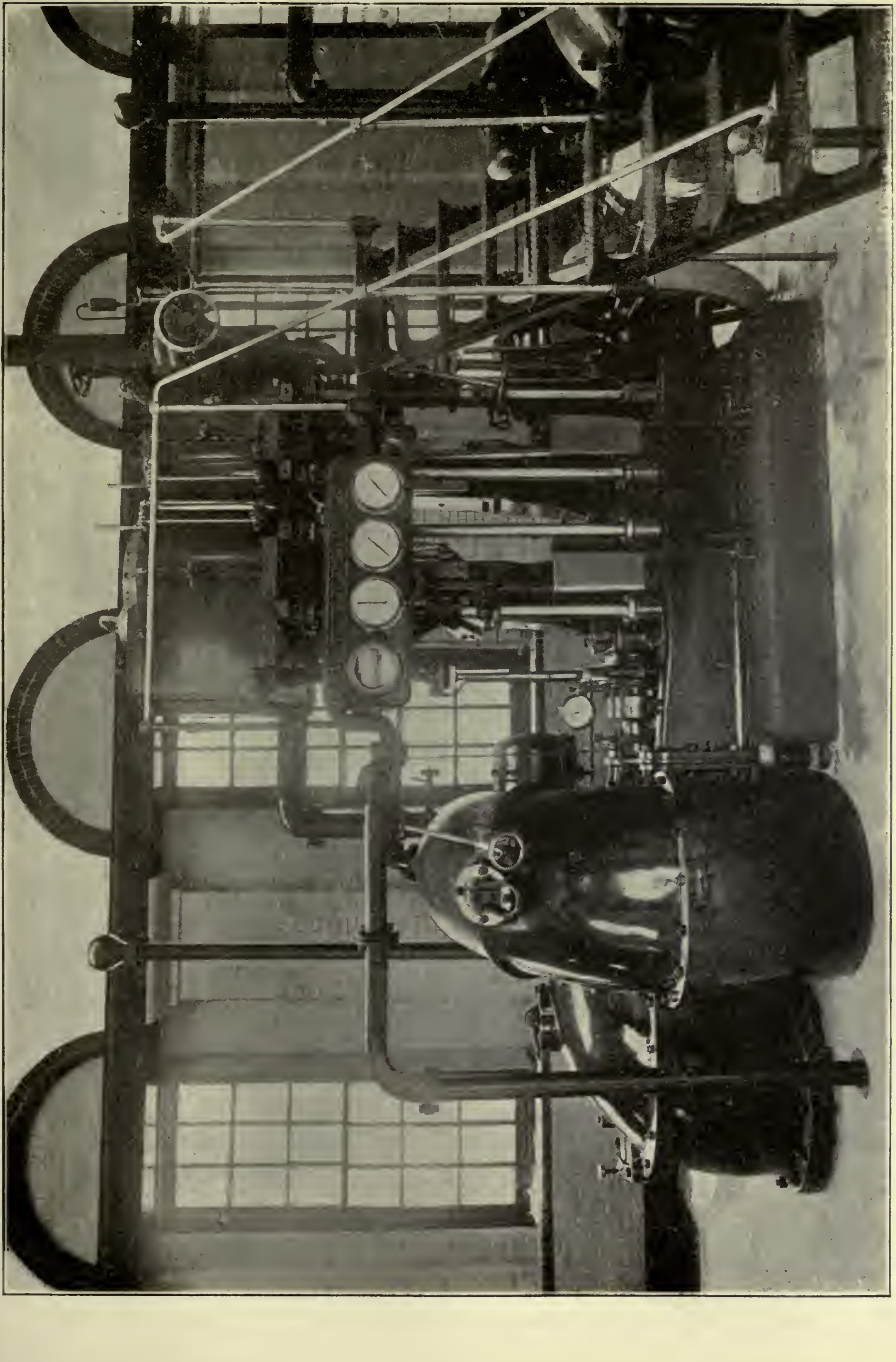


Fig. 251.—Centrifugal Pumping Steam Engine Plant for East London Water Works.

London Water Works, illustrated in Fig. 251, for filling their new reservoirs—Banbury and Lockwood—near the Angel Road Station, on the Great Eastern Railway. These reservoirs have a water area of 164¾ acres, and a capacity of 1198 gallons, and the brick-lined aqueducts for the inlet and outlet works extend over 1½ miles.

The pumps are capable of lifting 105 million gallons per day, and consist of three sets of centrifugal pumps, having 36 inches suction and delivery branches. Each set is driven direct by a coupled compound vertical engine, surface condensing, having 12½ inches high pressure, 25 inches low-pressure cylinders, with a stroke of 18 inches. The high-pressure cylinders are fitted with Meyer's variable expansion gear, adjustable outside by hand. The cylinders are lagged with mica composition 3½ inches deep, and covered with sheet steel.

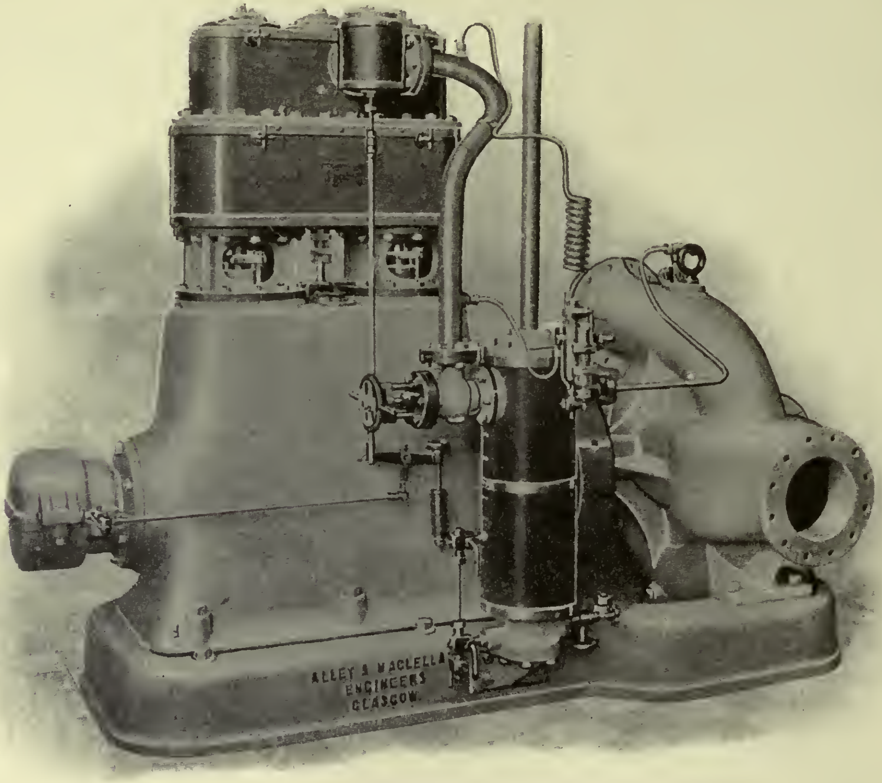


Fig. 252.—Alley & Maclellan Centrifugal Pumping Plant.

The surface condenser for each engine is fixed on the discharge branches of the centrifugal pumps, the delivery water flowing through the condenser tubes, which are of brass, 3 inches diameter. The air pumps are made in accordance with Edwards' patent, single-acting 9 inches diameter by 9-inch stroke, and are driven by rods connected to steel levers worked by links from the engine cross-head pins. The exhaust steam on its way to the condenser passes through Holden & Brooke's oil separators. A convenient platform with a staircase is provided in front of each engine for easy access to the sight-feed and other lubricators, indicator cocks, etc., but the engines can be worked either from the platform or ground line.

The steam is supplied by three Lancashire boilers, 7 feet 6 inches diameter by 30 feet long, at a working pressure of 140 lbs. One boiler will be kept as a stand-by. The steam pipes are of steel, lagged with mica composition.

The feed water is supplied by two horizontal treble ram pumps, one of which is also a stand-by; each set is driven through machine-cut gearing by a small 4-inch by 5-inch Tangye "Archer" engine, the exhausts from these small engines passing into a feed-water heater. The amount of water delivered into the boilers is measured by Kennedy meters. A traveller is fitted in the engine house, capable of lifting easily any part of the machinery. Each of the pumping engines Nos. 1, 2, and 3 was subjected to an exhaustive test for twenty-four hours, working under actual conditions, diagrams being taken every thirty minutes. The feed water was heated to 85° Fahr. The overflow from the air pumps and all drains was carefully weighed. Welsh coal was used, the weight being taken every hour.

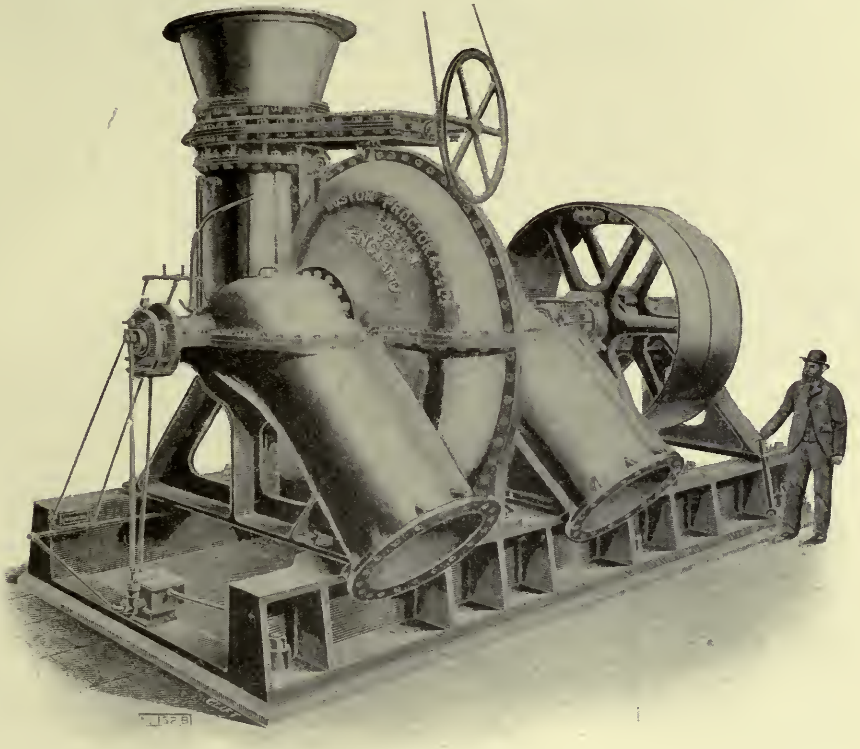


FIG. 253.—Centrifugal Ruston-Proctor Pump.

The engines, which are now at work, are capable, at a speed of 120 revolutions, of lifting 40 million gallons per day each into the reservoir.

In connection with centrifugal pumping engines the following account given by Mr. Livens of experiments carried out by him at Messrs. Ruston & Proctor's works on pumps are interesting. This well-known firm rank deservedly high as centrifugal pump makers. The following account is compiled from the contribution to the discussion on the efficiency of centrifugal pumps before the Institute of Mechanical Engineers made by Mr. Livens, 20th November 1903, a discussion well worth the attention of engineers interested in this subject.

The Ruston-Proctor pump for irrigation purposes is shown in Fig. 253. This pump, at 130 revolutions per minute, lifts 150 tons of water in that time, the wheel being 6 feet in diameter. This quantity of water at 10 feet head would give a water horse-power of about 100.

The experiments of Mr. Liven were made to ascertain the practical conditions for high efficiency. The theoretical efficiency is easily calculated both for turbines and pumps; but that facility is of no use whatever to the engineer who has to design actual pumps, a fact most strikingly proved by the pump makers being hitherto quite content with a 50 per cent. efficiency.

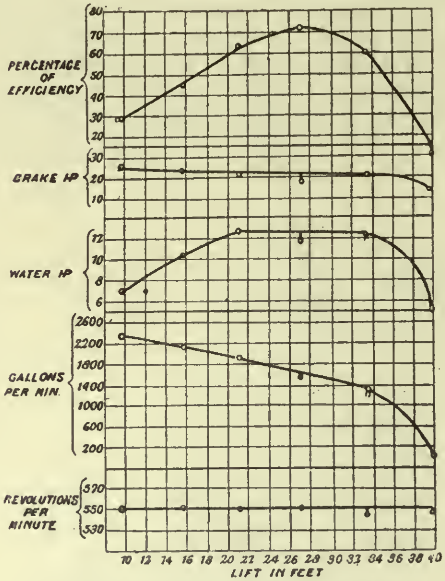


FIG. 254.—8-inch Centrifugal Pump. Constant Speed.

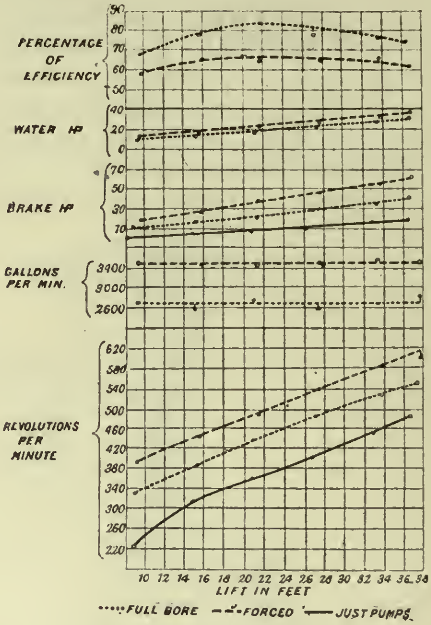


FIG. 255.—10-inch Centrifugal Pump. Test No. 1.

Referring to the diagram in Fig. 256, there were three forms of fans tried. The first illustrated a model pump in the works of Messrs. Ruston, Proctor, & Co. some years ago. The problem was to design certain large irrigation pumps, and model experiments were undertaken to throw light on the subject. The model illustrated had a fan 7½ inches in diameter. The results of the three forms of blades illustrated were shown in the diagram, Fig. 256. The fan blades *a* (Fig. 256), with the vanes well turned back, undoubtedly afforded the best form for general purposes, as shown by the efficiency curve. The design shown by *c* (Fig. 256), with the guide blades slightly turned back, gave better efficiency, the centre design *b*, with blades finishing radially, being inferior. During the experiments the maximum output was 230 gallons per minute, and when it was remembered that the fan was so small that it could be put in one's coat pocket the result would be allowed to be very satisfactory. As the result

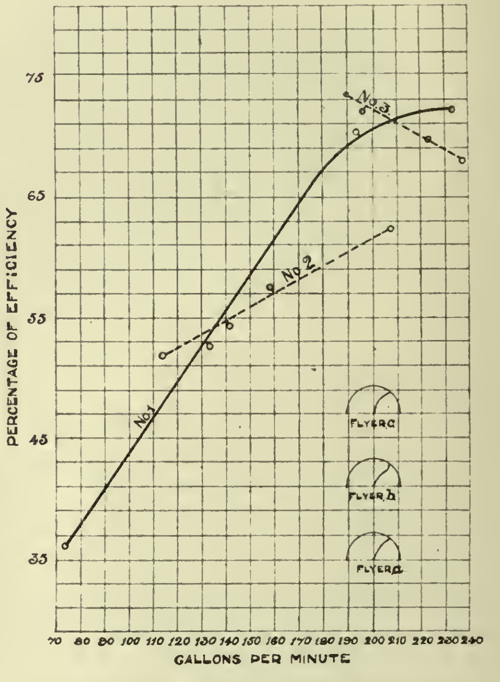


FIG. 256.—Diagram of 3-inch Model Pump. Lift 29 Feet.

of these experiments the large 36-inch pumps illustrated by Fig. 253 were designed. The lift was 27 feet to 28 feet, with an output of 135 tons per minute, the water horse-power being 66 per cent. of the indicated horse-power of the engine.

In designing the centrifugal pump it was necessary to take care that the water should enter in an easy manner, and the openings to the blades should therefore be of ample size. The cross section of the pump was of the greatest importance; in fact, it was of more importance than the shape of the blades themselves. With a fan of one of the forms illustrated the test shown in the diagram, Fig. 255, was obtained working normally; the efficiency was 85 per cent., and when the pump was forced it fell to 65 per cent. With a forced discharge the output was 3500 gallons per minute, the normal discharge being 2500 gallons per minute through a 10-inch pump, and the lifts, as shown by the diagram, were 10 feet to 36 feet. As to how these experiments were conducted, there was a large tank, holding over 12,000 gallons at a depth of 4 feet 6 inches. The capacity of this tank was carefully checked by means of a Kennedy positive meter at every 3 inches in depth. Each experiment in the pump took from one to three minutes. The pump was driven by a motor. The level of the water was taken both before and after the experiments, and the time of the experiments was that during which the switch which controlled the motor was open. The loss of time in the pump due to getting up its speed was nearly equal to the over-run in the pump after the current had been shut off. Before the experiments were made the efficiency of the motor was determined by a friction brake, and the friction of the motor was also arrived at; this is shown by one of the diagrams, Fig. 255. Experiments with the 8-inch pump gave results which are shown in another diagram, Fig. 254. It was run at a constant speed, and the lifts varied from 10 feet to 40 feet. At 40 feet the speed was just sufficient to keep the water held up in the pipe. At about 28 feet the efficiency of the pump was between 70 and 75 per cent. The quantity of water discharged varied from nothing at 40 feet until at 10 feet it was 2400 gallons per minute. The experiments were the more interesting as some persons

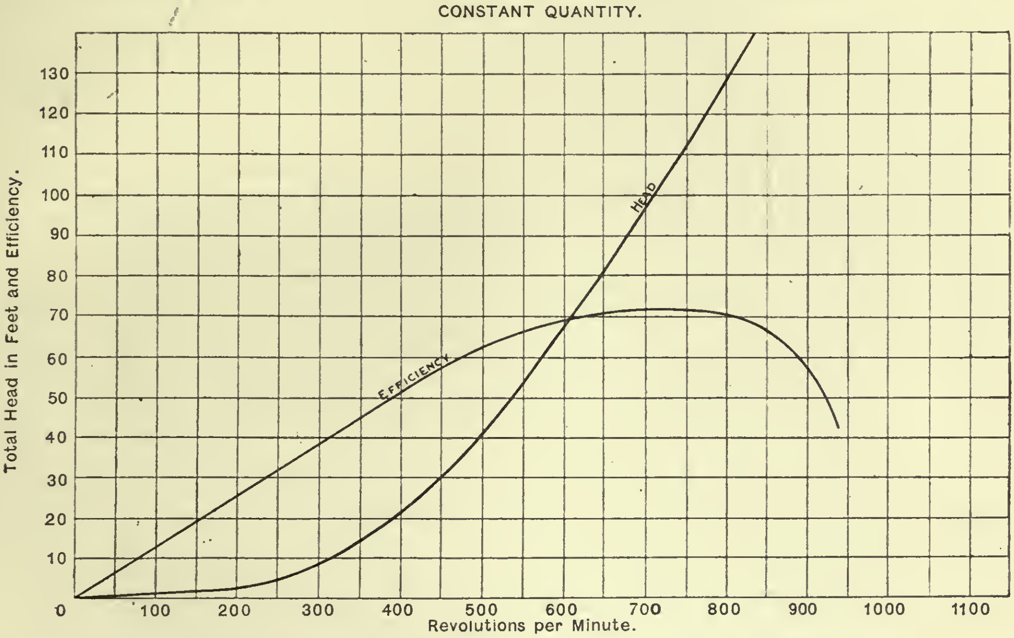


FIG. 257.—Curve showing Head and Efficiency at various Speeds at an Output of 1310 Gallons per Minute.

who use centrifugal pumps were under the impression that the smaller the height of lift the greater the efficiency; but within limits the more a pump was forced the better the result would be.

For low-lift pumps it seems quite possible in all sizes over 8 inches to obtain at least 80 per cent. efficiency.

In high-lift pumps the efficiency is not so satisfactory; but this applies equally to reciprocating piston and ram pumps. Messrs. Mather & Platt, who are the pioneers in this high-lift centrifugal pumping, give the two curves shown in Figs. 257 and 258, from which the performance of these pumps may be easily seen.

The author, in his experiments on marine propulsion by water jets, met with all the difficulties in the way of obtaining well-designed and efficient centrifugal pumps.

Many pumps are designed for the roughest of usage, and with little regard to efficiency. A pump giving one water horse-power will be found designed with enough cast iron and dimensions as if it were to be subjected to enormous hydraulic strains. At the present moment, if a pump of high efficiency is required, and to be designed with all its component parts proportional to the forces and strains, and a sufficient factor of safety is required, it has to be specially designed and made.

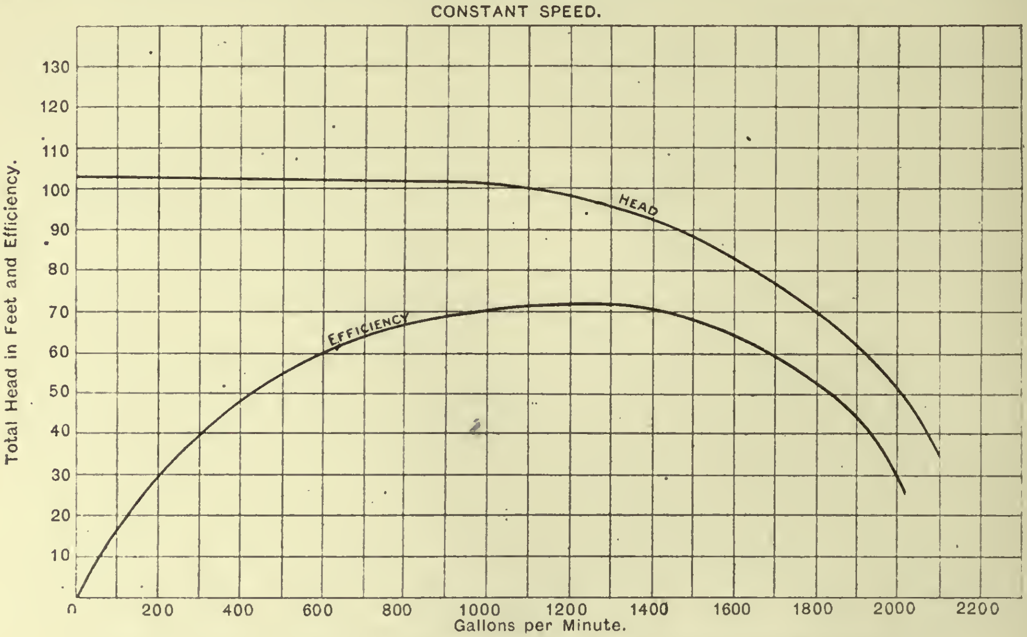


FIG. 258.—Curves showing Head and Efficiency at various Outputs at 700 Revolutions.

SCREW PUMPS

Fig. 259 illustrates what is probably the most efficient machine ever devised for handling large quantities of water under very low heads. This engine was built for the city of Milwaukee, and is employed in forcing clear water from Lake Michigan through a 12-foot diameter tunnel, 2500 feet long, into the Milwaukee River, for flushing purposes during the summer months.

This machine was designed by Mr. Edwin Reynolds, to whom is due the credit of originating a type of pumping engine for discharging large quantities of water under light heads, that for this service approaches very near to plunger or piston pumps in efficiency. This type performs about double the amount of work with the same fuel, and costs one-half the price asked for centrifugal pumps adapted to the same service.

The pump is a modified form of propeller wheel with screw-shape blades. This is mounted in a cast-iron circular housing or casing, which is set in the brick walls of the tunnel. The wheel is keyed to the crank shaft of a vertical compound engine. Suitable bearings for supporting the outer end of the crank shaft are formed in the wheel casing, these being fitted with devices for adjusting the journals and keeping the shaft in line.

The engine is of the Reynolds vertical compound condensing type, with steam-jacketed cylinders, attached air pump and jet condenser. It is set in a boiler-iron pit enough below the intake level, so that the wheel is always submerged whether it is running or at rest.

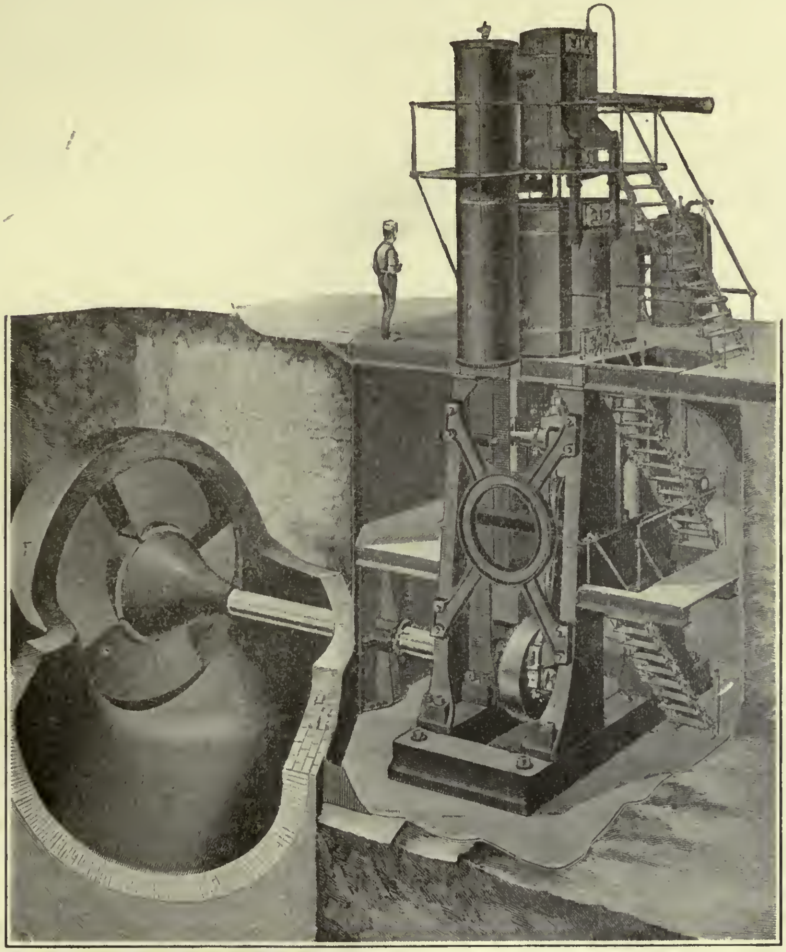


FIG. 259.—Screw Pump for Drainage.

The contract capacity was 32,000 cubic feet per minute, raised $3\frac{1}{2}$ feet high, but the engine is ordinarily run at a capacity of 40,000 cubic feet per minute, and is easily capable of delivering 48,000 cubic feet per minute, or over 500,000,000 gallons in twenty-four hours. As far as can be ascertained, this is the greatest quantity of water handled by any single machine in the world. The guaranteed duty was 70,000,000 foot-lbs. of work delivered for each 100 lbs. of coal burned, without deductions of any kind, and the duty shown by the official test was 76,000,000 foot-lbs., and the average yearly duty on all coal burned is fully equal to that shown by the official twenty-four hour test.

Machines of this type may be built with various types of engines, simple, compound, or triple-expansion, or may be arranged to be driven by variable speed electric motors—three electric driven screw pumps having been installed for the New Orleans Drainage Commission. For the city of Chicago two screw pumps driven by vertical triple-expansion engines to pump against a 7-foot head and to deliver 40,000 cubic feet per minute are working.

MINE PUMPS

The early mining pumps nearly all consisted of beams, operating pumps in the pit or mine shaft by pump rods hung direct from the beam and extending down the shaft.

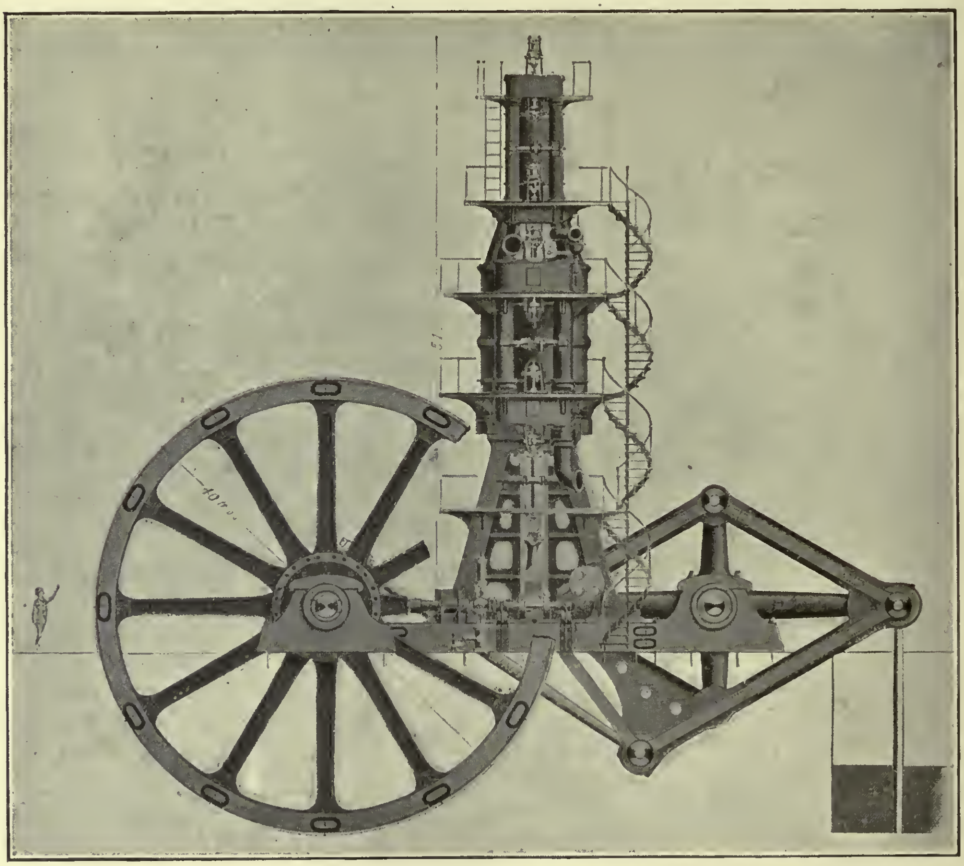


Fig. 260.—Bell Crank Mining Pump.

The mechanism was necessarily very heavy, and speeds very slow. At the present day these engines would be outclassed entirely by centrifugal working in two or three stages by electromotors.

The most modern design of this old type of large size is shown in Fig. 260. The bell crank is oscillated directly by the connecting rod of the engine, while a long connecting rod transmits motion from the lower centre of the bell crank to the fly-wheel. To the overhanging outer end of the connecting rod the pump rods are suspended. The engine is a compound vertical tandem engine.

The illustration (Fig. 260) shows a large pumping engine for the Chapin Mining Company. The machine is designed to pump 3000 gallons per minute from a depth of 1500 feet. The illustration will give some idea of its proportions.

The high and low-pressure steam cylinders are 50 and 100 inches diameter respectively, and have a common stroke of 10 feet. The fly-wheel is 40 feet in diameter, and alone weighs 164 tons; the total weight of the engine, exclusive of pumps and shaft work, being 600 tons.

The pump rod is attached directly to the beam end, which overhangs the shaft. The pumps are of the single-acting Cornish type, with plungers 28 inches diameter by 10 feet stroke; the number of strokes per minute varying from 4 to 10.

The pumps are arranged in series, each set having a lift of about 200 feet. Modifications of this type of machine are made designed to suit any requirements.

TRIPLE-EXPANSION PUMPING ENGINES

An American design of a pumping engine working on the same principle as that referred to in Fig. 250 is shown in Fig. 261.

The general features of the design are clearly shown in the illustration, and, as will be seen, the arrangement of the engine above the floor is similar to the marine engine known as the "fore-and-aft" type.

There are three steam cylinders, "high," "intermediate," and "low," each cylinder having its piston rod connecting to a steel cross-head, and from this cross-head extend four steel tie rods to the plungers, thus forming a direct and rigid connection between the steam pistons and pump plungers.

The main shaft has three cranks set at angles of 120 degrees from each other, which gives a steady turning effect, and as the pump plungers move in unison with the cranks a perfectly uniform flow of water is obtained, reducing the shock and jar on pumps and pipes to a minimum.

The cylinders are steam jacketed, covered with non-conducting material and lagged with steel.

The cylinders are fitted with Reynolds Corliss valve gear, especially arranged for this type of engine, adjustable trip cut-offs being provided for each cylinder; the high pressure valves being also controlled by a speed governor.

The throttle valve, injection valve for condenser, cut-off levers, etc., are all placed conveniently, so that the engine can be stopped, started, and controlled from one position on the engine-room floor.

The pumps, which are located below the engine bed plates, are of the single-acting, outside packed, plunger type, and are provided with large valve area, ensuring quiet working at high speeds.

The pump valves are of rubber, and are arranged in groups mounted on cages, which are easily removable, manholes being provided giving easy access to the interior of the pumps.

The pump chambers are circular, this form giving the greatest possible strength. Especial care is taken in designing the pumps to avoid shrinkage strains in the castings. The valve chambers support the engine bed-plates in whole or part, this arrangement reducing the amount of foundation required, and at the same time rendering the pumps much more accessible.

The pumps can be placed any required distance below the engine-room floor by simply extending the plunger connections. The air and feed pumps are driven direct from the main engine, this making a more economical arrangement than an independent condensing apparatus. The condenser may be either of the surface or jet type, as local conditions require.

In regard to economy, we would say that there is an erroneous impression that nothing is gained by using a triple-expansion engine with steam pressures less than 160 lbs. Experience has shown that with steam at any pressure above 80 lbs. the triple-expansion engine will show an economy from 10 to 15 per cent. better than a compound engine of the same general construction using steam at the same pressure. And 75 per cent. of the triple-expansion engines built are operated with steam pressures of 125 lbs. or less, the lowest being 80 lbs.

With triple-expansion engines using steam at from 90 to 110 lbs., duties of 120 to 140 millions may be obtained, while with steam pressures at from 125 to 150 lbs. a triple-expansion engine will run up 140 to 170 millions; and very probably with steam pressures at from 175 to 200 lbs. under favourable conditions, the triple will give above 175 millions duty per 1000 lbs. of steam.

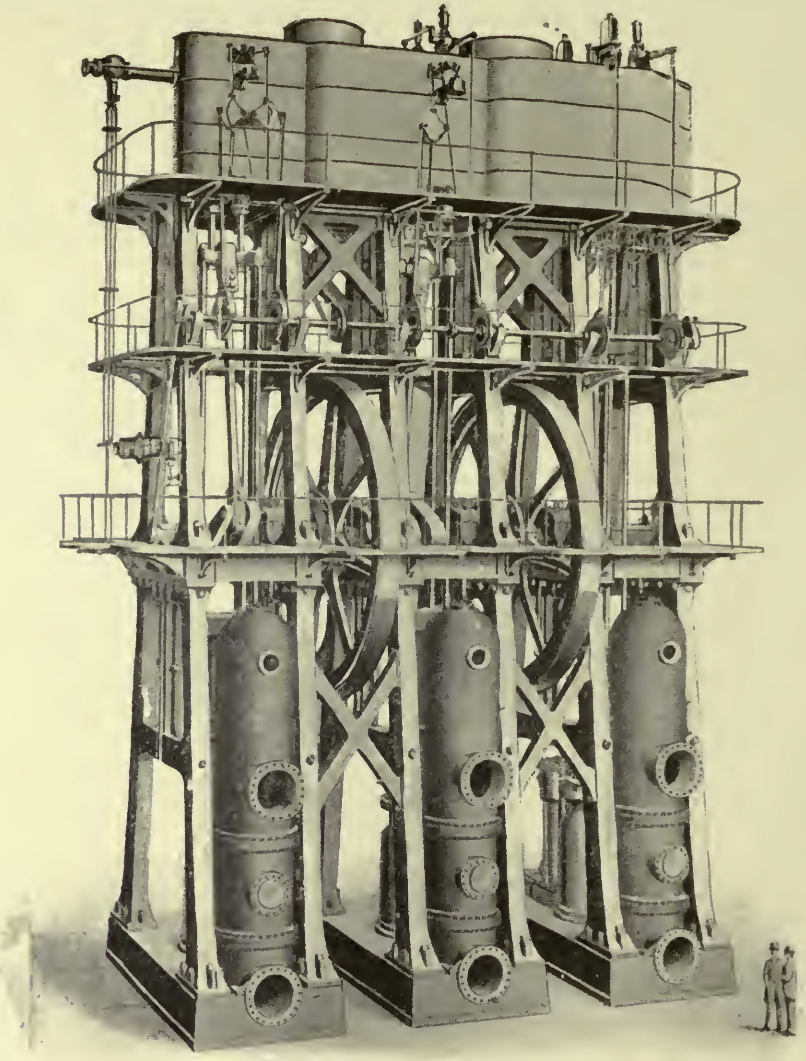


Fig. 261.—Triple-Expansion Ram Pumps.

The world's record of economy (154 millions duty,—1168 lbs. of steam per indicated horse-power) was held for several years by the 18 million gallon triple-expansion engine of the Milwaukee Water Works, this engine using steam at only 125 lbs. pressure.

SMALL WATER PUMPING ENGINE

The pump shown in Fig. 262, commonly known as a pendulum pump, is a very common type. The pumping is direct between the steam and water cylinders, the connecting rod, crank shaft, and bearing have only their own weight and a small strain due to the fly-wheel carrying over at the dead centres to transmit.

VARIOUS TYPES OF PUMPS

The different types of centrifugal pumps have been discussed in Volume I. The reciprocating types of pumps employed in connection with reciprocating engines require some attention. We have already referred to the slow speeds required for working reciprocating pumps. The piston speeds in pumps vary from 60 to 200 feet per minute. The former for short-stroke pumps, and the latter for long-stroke pumps. Higher piston speeds have been successfully used against low heads in large pumping engines. Large valves and large pipes are very essential in such cases.

The frictional losses in valves increase very rapidly with the speed of the water. At and under 3 feet per second the losses are proportional to the speed, but above 5 feet per second it increases to the square of the speed, and at higher velocities it is proportional to more than the square of the speed.

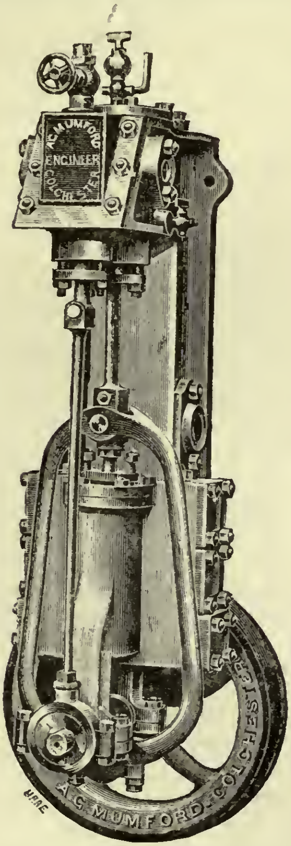


Fig. 262.—Donkey Pump.

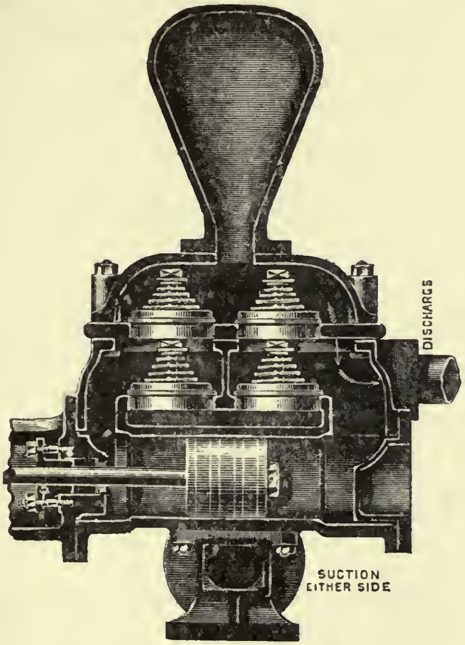


Fig. 263.—Application of Air Vessel.

In order to steady the flow of water in the pipes it is necessary to provide on the suction pipe, if it is long, a suction chamber. And on the delivery side of the pump an air vessel to provide a spring which takes up the surplus water at the maximum thrust of the pump, and returns at the minimum thrust or dead points. This variation of thrust is negligible in direct acting slow-speed pumps, as the speed throughout the stroke is uniform; but in pumps driven by a crank the speed varies throughout the whole stroke. In the first class of pump the air vessel is unnecessary; in the latter it is much needed.

When a suction chamber or pipe is required it should be placed so that the water enters it straight, without requiring to bend or deviate.

As to the air vessel, it is usually fitted to the delivery chamber of the pump, as shown in Fig. 263. Care should be taken to see that it performs its functions, and that it really does contain air. It should have means for replenishing it with air, and also for indicating the height of water in it.

Different types of pumps may be found in the sheet of diagrams, Fig. 264.

No. 1 is the common single-acting suction pump, with a valve on the piston and a valve on the inlet.

No. 2 is a diagram of a single-acting ram force pump.

No. 3 is a diagram of a double-acting ram pump externally packed.

No. 4 is a diagram of a double-acting ram and piston pump, forces on both strokes, sucks only on the up-stroke.

No. 5 is a diagram of a double-acting piston pump with four valves, the common form of feed pump.

No. 6 is a combined bucket and plunger pump, sucks on the up-stroke only, but delivers on both strokes.

No. 7 is a diagram of a double piston pump. The water is pressed by both pistons. Each piston has a valve opening opposite ways.

No. 8 is a diagram of a double-ended ram pump.

BLAST FURNACE ENGINES AND AIR COMPRESSING ENGINES

In the smelting of iron and in the production of steel and other metallurgical operations a blast of air is required, and to supply this various classes of blowing engines are in use. And for many mechanical purposes compressed air is required.

For low pressures up to about ½ lb. per square inch the fan is the best blower. We have already described the fans in use in Volume I.

For pressures above that reached by fans, and not above 1 or 2 lbs. per square inch, the Roots blower is used. This blower acts upon the displacement principle by means of two oblong rotary pistons, shaped as shown in the diagram, Fig. 265. The two pistons are on parallel shafts geared together by toothed wheels at one end, and driven by power at the other end.

The diagram, Fig. 266, shows two steam cylinders connected to the pistons for driving them. It is a crude machine, but serves for the purpose fairly well. The blast produced is not steady, and the gear wheels always develop a large amount of back lash, so that a good deal of power is wasted in producing vibration and noise.

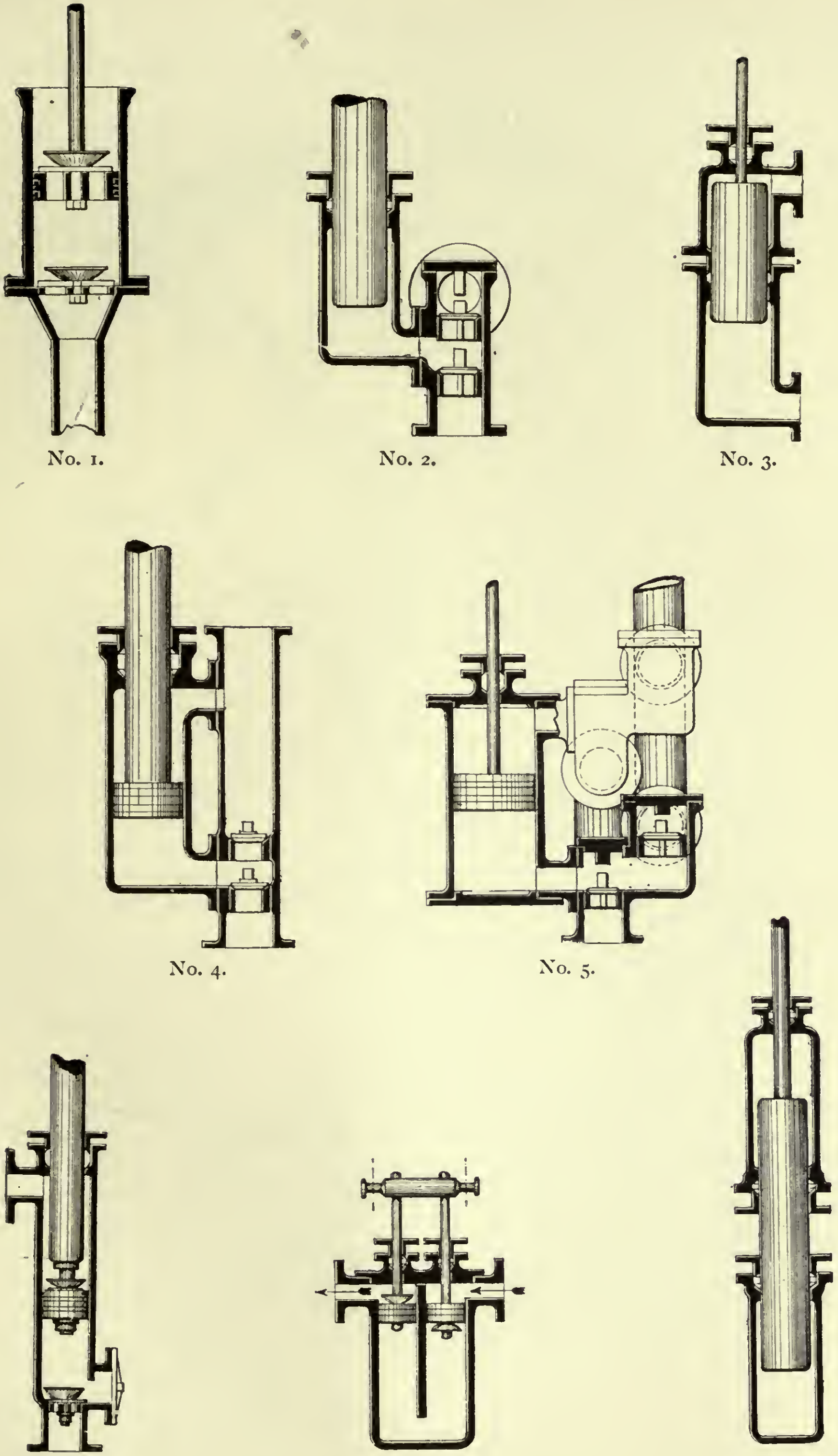
Mr. Parsons' screw air compressor is a great improvement upon this, and has already been described in Volume I.

Boilers are sometimes urged to greater activity by artificial draught, produced by fan blowers driven by engines. There are two systems, the forced draught and the induced draught. In the former the air is forced into closed furnaces below the fire bars, or, in case only a small additional pressure is required, sometimes the stokehole is closed and the air maintained under pressure by a fan.

In the induced draught system the fan is placed with its outlet in the chimney, and its inlet drawing the furnace gases from the boiler flues.

The forced draught system with closed ash pits is the best, as it is easier regulated, and the inrush of cold air is less when the fire doors are opened than with induced draught.

Fig. 267 shows an end and side view of a water tube, Babcock & Wilcox boiler



No. 1. No. 2. No. 3.

No. 4. No. 5.

No. 6. No. 7. No. 8.

Fig. 264.—Types of Pumps.

partly in section, showing the arrangement of the fan and flues, and Fig. 268 shows a blowing engine and fan by Alldays & Onions.

For mine ventilation fans are universally employed. The old style was to employ fans of very large diameter, in order to get the high peripheral speeds required to give the air the centrifugal impetus.

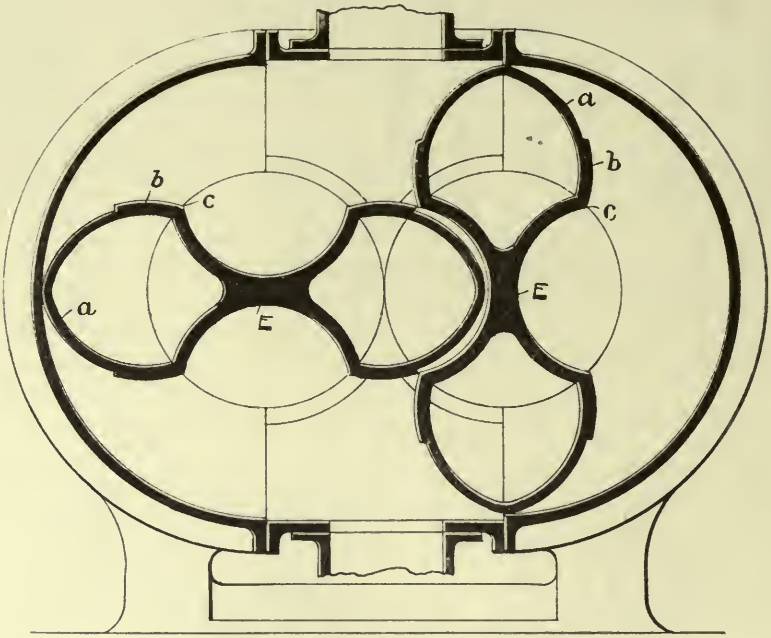


Fig. 265.—Roots Blower.

The perfection of high-speed engines and turbines has, however, rendered these large wheels unnecessary, and quite a small wheel at a high number of revolutions is now the vogue.

Fans for high speed should have very short curved blades. The blades should be thin and numerous. These conditions are met with in the Sirocco fan.

A high class engine and fan by Messrs. Alldays & Onions, Birmingham, is shown in Fig. 269, fitted to a compound high-speed engine. The combination takes up little space, and is highly efficient.

The piston and cylinder air blowers are used of immense dimensions for blast furnaces, and at a not far distant date were universally steam driven, and mostly took the form of beam engines. But in some cases the design was more modern, the air cylinder and steam cylinder being in line, and the power thus transmitted directly from piston to piston, without acting through joints and cranks and rods. The fly-wheel and crank is introduced to regulate the motions and operate the valves only. An engine for blowing on this principle is shown in Plate IV.

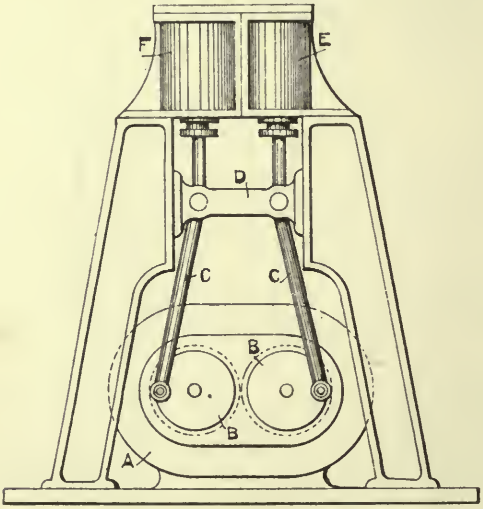


Fig. 266.—Roots Blower.

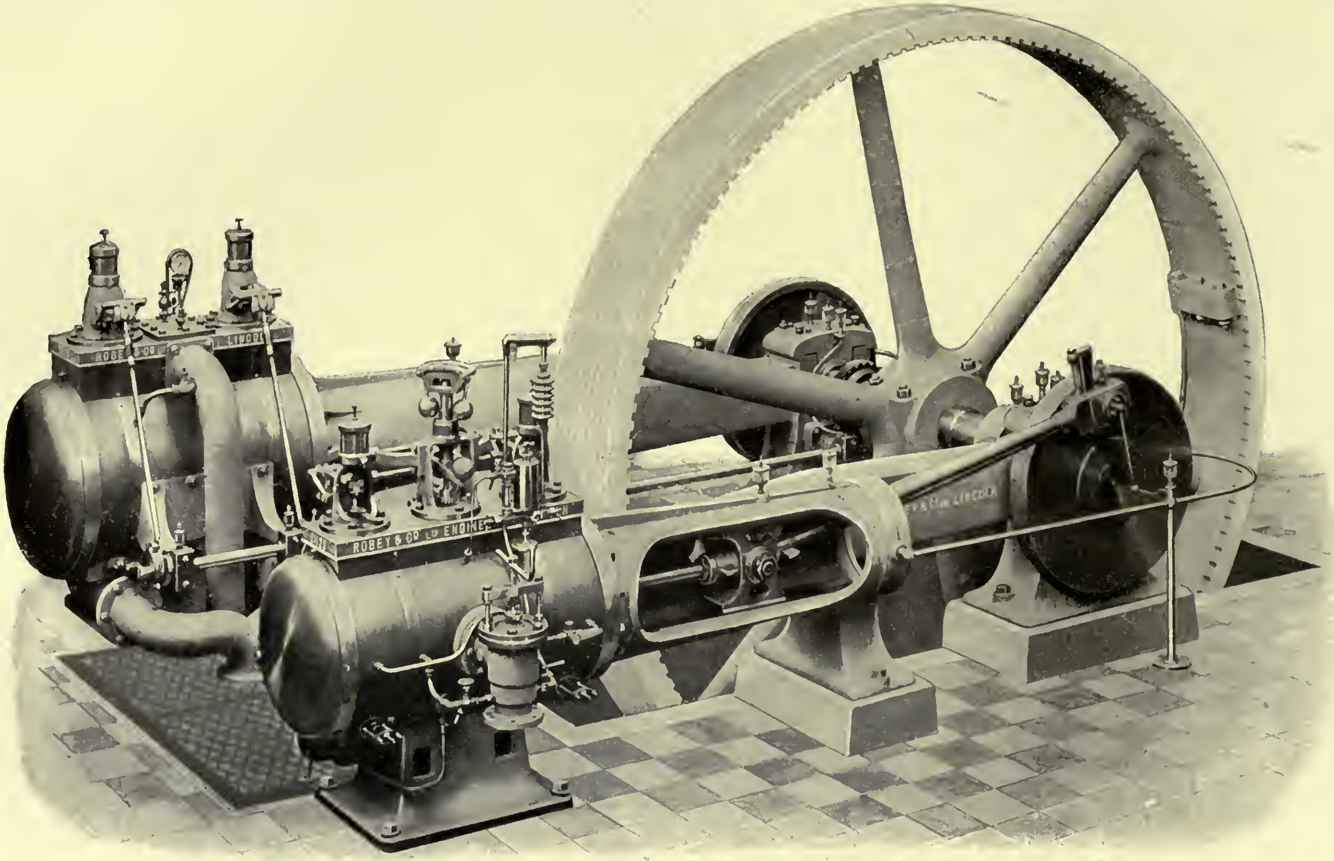

ROBEY & CO
LINCOLN
ROBEY & CO LD ENGINE

PLATE VI.—HORIZONTAL COMPOUND DROP VALVE MILL ENGINE.

VERTICAL STEAM BLOWING ENGINE

A fine example of a modern vertical engine has just been erected, and the makers have given the details of its dimensions as under. The full-page illustration (Plate IV.) shows its general design somewhat of the marine engine type. It is in operation at the Clarence Works, and was manufactured by Messrs. Richardsons, Westgarth, & Co., of Middlesbrough. Each pair of engines is separate and self-contained, with steam cylinders 48 inches and 84 inches in diameter by 60-inch stroke, fitted with Corliss valves; those for the high-pressure cylinder being controlled by a Whitehead patent governor, with gear so arranged that the engines can be set to run at any desired speed. The air cylinders are 84 inches in diameter by 60-inch stroke, fitted with a series of circular valves at each end. One of the engines is fitted with ordinary metallic valves, and the other with Westgarth's patent pneumatically cushioned valves. The engines are designed to work with steam at 90 lbs. pressure, and to blow air up to 20 lbs. pressure, the normal working speed being 35 to 40 revolutions, but the engines are capable of running continuously up to 50 revolutions per minute. The general construction of the engines and framing, etc., are well shown by the engraving. The bedplates are 4 feet deep, and exceptionally heavy. The crank shaft is 18 inches in diameter at the bearing, 22½ inches diameter for the flywheel; crank pins, 12 inches in diameter and 15 inches long; fly-wheel, 16 feet in diameter and 35 tons weight. The total weight of each pair of engines is about 310 tons. There is a steam-jacketed reheater between the high-pressure and intermediate steam cylinders; and, as shown by Plate IV.,

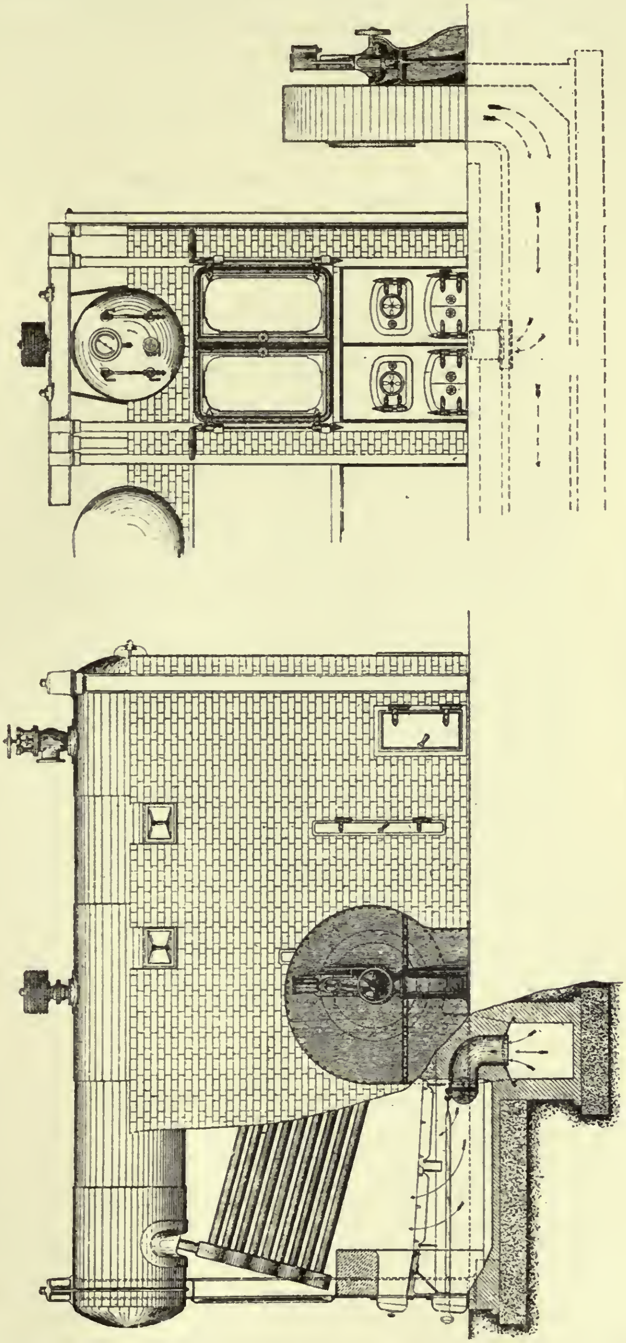


FIG. 267.—Forced Draught.

there is a very complete arrangement of platforms to give access to all working parts.

The air blowing engines with reciprocating pistons is a most important engine in many industries, and much has been accomplished by engineers to render them highly efficient. In Paris, Professor Riedler has brought the system of compressing air in stages with intermediate cooling to a practical result in the compressed air transmission system laid out for public supply in that city. The improvements effected from time to time, recorded at the Station Rue St. Fargeau, made upon the blowing engines used there are very significant. In the original plant the quantity of air compressed to 90 lbs. pressure was 20.25 lbs. per indicated horse-power. These first compressors were substituted by Davey-Paxman compound engines, each driving two compressing cylinders of 2000 horse-power in all. The compressors were of Swiss make by Blanchard. The result was 22.92 lbs. of air compressed to 90 lbs. per indicated horse-power. Professor Riedler converted one of the Cockerill compressors to his system of compounding, which then gave 28.27 lbs. of air at 90 lbs. pressure per indicated horse-power. Continuing the improvements, new compressors were made, with the result of 33.47 lbs. of air at 90 lbs. per indicated horse-power. The final result of all the improvements being 67 lbs. of air compressed for one penny.

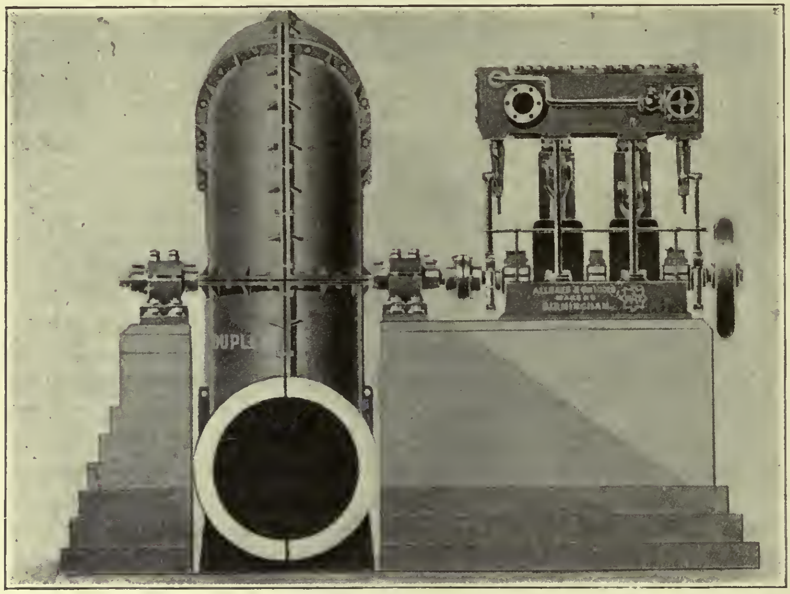


Fig. 268.—Ventilating Fan.

These results are very interesting, and give an idea as to the cost of compressed air with modern engines.

In blowing engines for furnaces and iron and steel work generally less than 20 lbs. pressure is sufficient, but the volume of air required is very large. Valves and passages are therefore very large in proportion to the air cylinders. An improvement consists in arranging the valves to be opened and shut mechanically, instead of by the pressure and suction of the air.

A number of blowing engines worked by the gases from the furnaces (see page 199, Fig. 281) is a striking picture of modern progress, and is a credit to engineering enterprise. This is the sort of progress which makes for successful competition: not by sitting still, grumbling about the bad trade in old played out engines and machinery; but by boldly taking up the most recent developments and placing them on the market as early as possible. The enterprising engineer has no reason to fear any competition, protected or unprotected.

In some parts of the world there are many minerals and coal, or plenty of timber, but not much water for steam purposes. These engines meet this case by gas fuel produced from coal and wood.

Twenty years ago the blast furnaces burned all the gases made in the process in one large flame at the furnace top, absolutely wasting them. The growing competition in the iron trade compelled savings to be made wherever possible. And the utilisation of the furnace gases was among the first proposals. Nowadays the gases are used to produce valuable chemicals, and also all the power required for the works.

A very good type of piston air compressor is shown in Fig. 270 partly in section, and in Fig. 271 complete.

The "Reavell" compressor has four cylinders arranged radially in a circular-shaped casing, as shown in the sectional illustration, Fig. 270.

Each of the four cylinders is fitted with a trunk piston; and the four connecting rods are all driven by a common crank pin. The casing contains an annular space through which the cylinders pass, and which is used as a water jacket. Each cylinder forms, as it were, a separate single-acting compressor, and as they all deliver into a common delivery passage, a practically continuous delivery of air is secured, seeing that the compressor, owing to the pressure being always in one direction, can be run at relatively high speeds.

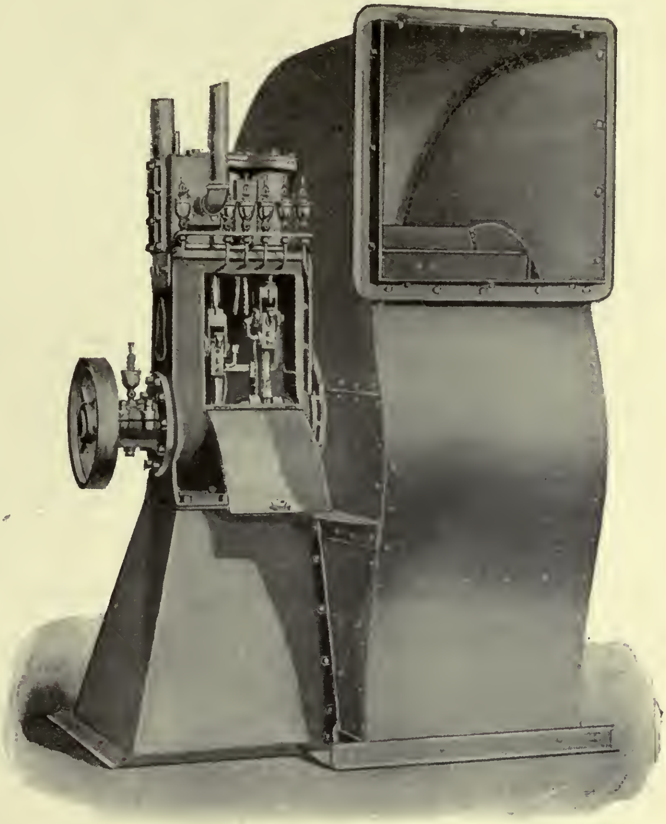


Fig. 269.—Ventilating Fan.

The compressor has no suction valves, air being admitted above each piston by means of a port in the latter, which coincides with a similar port in the top of each connecting rod, during the suction stroke; and near the end of this stroke the piston overruns the ports cut through the cylinder wall, as shown, thus making direct communication between the cylinder and the inside of the compressor casing, which is arranged to form a suction chamber.

Delivery valves are fitted at the outer end of each cylinder, which open during the compressing stroke as soon as the air has reached the required delivery pressure, and through which the air passes to the delivery belt or passage shown, from which it may pass through any of the four outlets provided.

The sectional views will make the construction clear. The section through the shaft and bearing shows the circumferential delivery port. The section through the cylinders shows a small balance piston, which is fitted to the larger sizes in order to obtain constant thrust; the special feature of this compressor ensuring silent working.

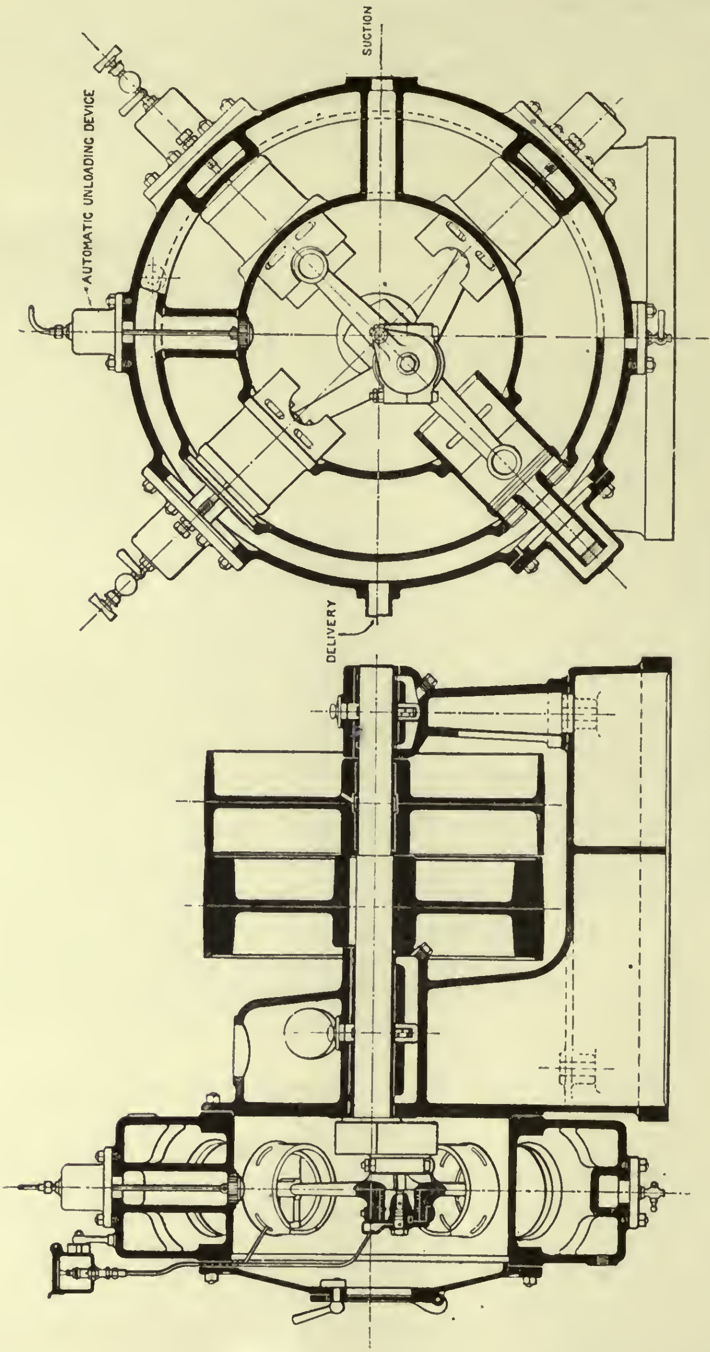


FIG. 270.—Reavell Air Compressor.

The door enclosing the suction chamber is of sufficient size to enable the whole of the working parts to be readily examined or removed.

The connecting rods (Fig. 272) are so designed that they can be withdrawn sideways from the piston when the latter is at the end of its stroke, and it is possible to remove the whole of the working parts of the machine without breaking any pipe joints.

The photographs show the hole in the side of the piston for the removal of the connecting rod. The suction ports in the piston and connecting rod gudgeon are also clearly shown.

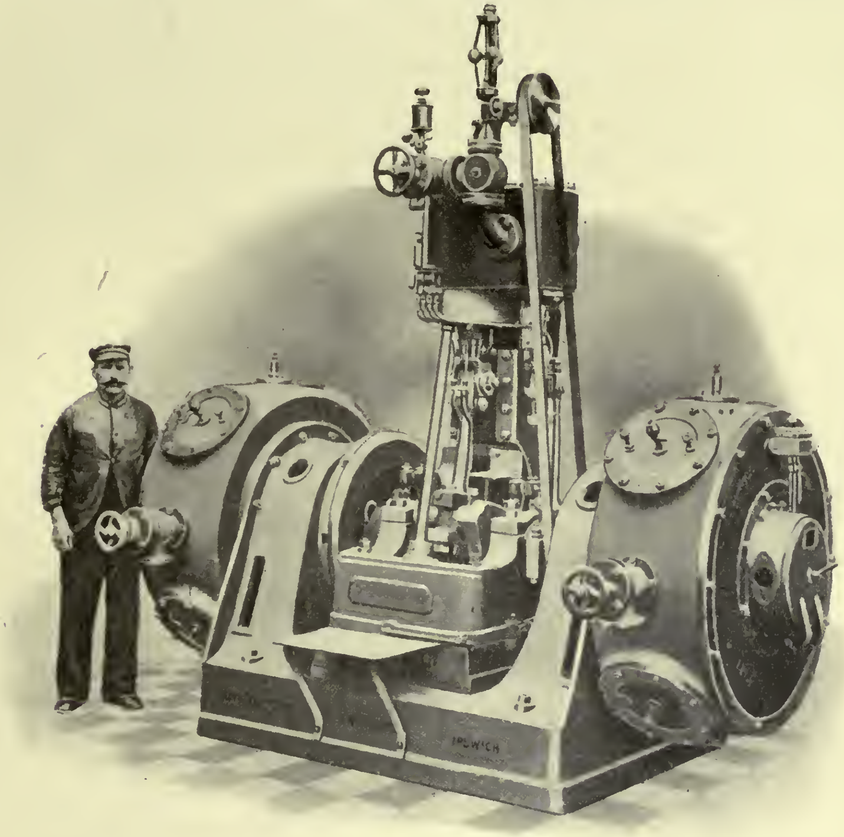


FIG. 271.—Reavell Air Compressor.

The machine may be thoroughly dissected for examination within five minutes.

The sectional view (Fig. 270) shows a machine of the belt-driven type, with fast and loose pulleys.

The diameter of the driving pulley is such that the machine can be readily driven from a line shaft pulley, without the intervention of a counter shaft to increase the speed, as is needed with other compressors.

A loose pulley is provided on the shaft of the machine itself.

An automatic unloading device can be provided with this compressor. This device is in the form of a bypass valve, forming a connection between the delivery and the suction side of the compressor. It is automatically controlled by an air relay, so that when the air pressure reaches the desired limit the bypass valve opens, and the delivery chamber is put in communication with the suction side of the compressor.

FIG. 272.

The load is thus removed from the driving pulleys, or, if the machine be arranged for electric driving, from the motor.

As soon as the pressure falls to a predetermined limit this valve closes, and the compressor commences to deliver air again.

This bypass arrangement is shown on the sectional view.

THE QUARTER CRANK BLOWING ENGINE

This engine, made by Richardsons, Westgarth, & Co., is shown in Plate V.

The steam cylinders are compound, 42 and 84 inches diameter; the air cylinders are 84 inches, all 60-inch stroke, the steam pressure being 125 lbs. per square inch, exhausting into the condenser. The ordinary speed of the engine is 45 revolutions, and will deliver air at 12 lbs. pressure. They are, however, capable of running up to 60 revolutions per minute, and of blowing 20 lbs. when required.

One set of these engines has been running for over a year at the Barrow Company's Works, and, we believe, giving every satisfaction. Another of these engines is now erected at the Frodingham Company's Works, near Doncaster, and a low-pressure side only has been put down at the Ebb Vale Steel and Iron Company's Works, using the exhaust steam from several of their other engines. This engine has been running for nearly a year now.

One important feature is the air valves known as the Southwark valves. The valves are of the gridiron pattern, one inlet and one outlet valve at top and bottom of cylinder, as will be seen from the drawing (Fig. 273). The clearance is very small, being less than 2 per cent. When it is remembered that the original engines (many of which are still working) in this district have a clearance of 20 per cent., it is evident what a great advantage the Southwark valve possesses, as the inlet and outlet orifices of the valve are respectively 13 per cent. and 10 per cent. of the cylinder area, so this has not been sacrificed to get small clearance. Fig. 273 shows this type of valve in detail. The inlet valves are controlled by means of cams and levers receiving their motion from an eccentric on the main shaft, while the outlet valves are opened automatically by means of a piston on the actuating stem of the valve upon which the pressure in the air cylinder acts. As showing that the Southwark Foundry Company are alive to the importance of having the most effective and simple valve, in 1899 the outlet valve was closed by means of letting in the air from the main to behind the piston of the actuating cylinder; this was effected by a positively worked piston valve, but now the outlet valve is positively closed at the end of the stroke by a cam. The opening of the outlet valve is automatic and opens widely, and instantly the air pressure in the cylinder is equal to that in the receiver. The valves before moving lift from their seats, so that there is neither friction nor wear.

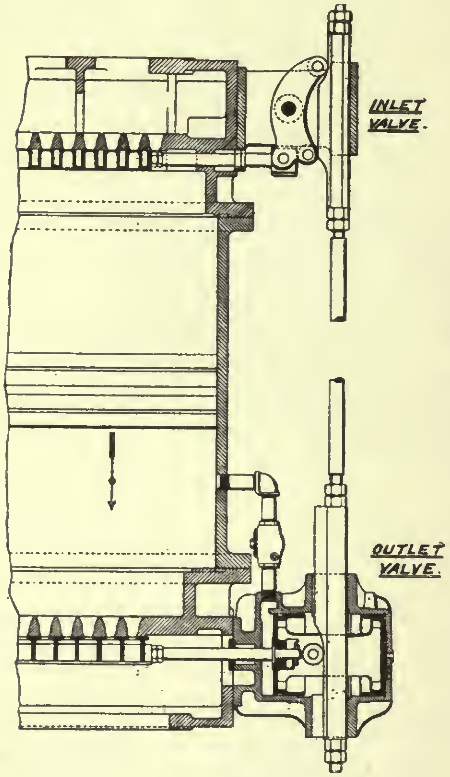


FIG. 273.—Valves of Quarter Crank Engine.

The pumping and compressing of gases involve many difficult problems. Inefficiency arises from clearances, from heat of compression, leakage, and other causes, all of which will be considered under Power Transmission.

MILL AND FACTORY ENGINES

As an example of a modern form of this class the compound horizontal Robey engine, shown in Plate VI., is selected, especially adapted for large industrial works and central electric light installations. The steam admission valves are of the double beat equilibrium type, their fall on to their seats being softened by dashpots. The lifting wipers are driven by eccentrics on a shaft turned by skew gear; the depth of hold of the wipers is controlled by the governor, which gives a horizontal motion through a small range to the bearings carrying the fulcrum bar of the wipers. The gearing works without noise, and the engine could be stopped in case of emergency by throwing up the wipers out of gear by hand

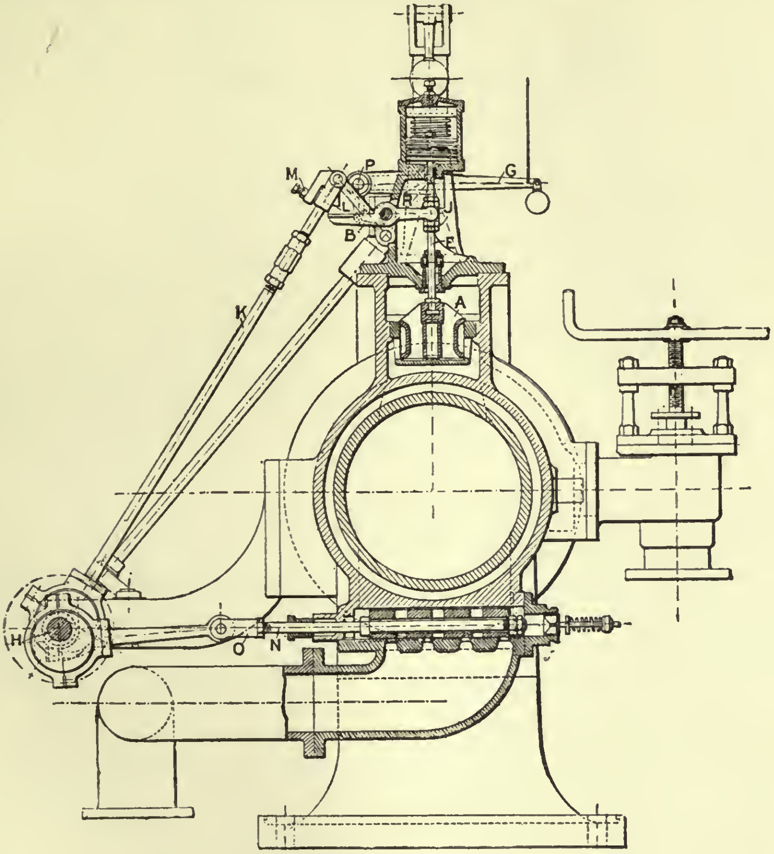


Fig. 274.—Robey Drop Valve Gear. Cross Section.

There is a steam-heated intermediate receiver or hotpot between the cylinders and under the floor. All the working parts are very large. The main shaft is swelled in the middle.

Besides for industrial purposes, large numbers of these engines have been used for central electric light installations, amongst which are that of the Newcastle Electric Supply Company, the lighting of Bilbao, the city of Melbourne, and many others.

From very careful indications taken at the first named of these it was found that the maximum variation of speed has never reached one revolution per minute with any variation of load.

The Robey type of drop valve and trip gear is shown in sections (Figs. 274 and 275).

In the longitudinal section AA are the two admission valves, and, being almost in equilibrium, are lifted with great ease by the small eccentric and rods worked from a shaft revolving at the same speed as the engine, and parallel with the bed-plate.

The admission valve is raised at the commencement of the stroke, and held wide open until the point of cut-off is reached, when it is instantly relieved and falls on its seat, an air cushion in the guide cylinder above preventing its falling too heavily.

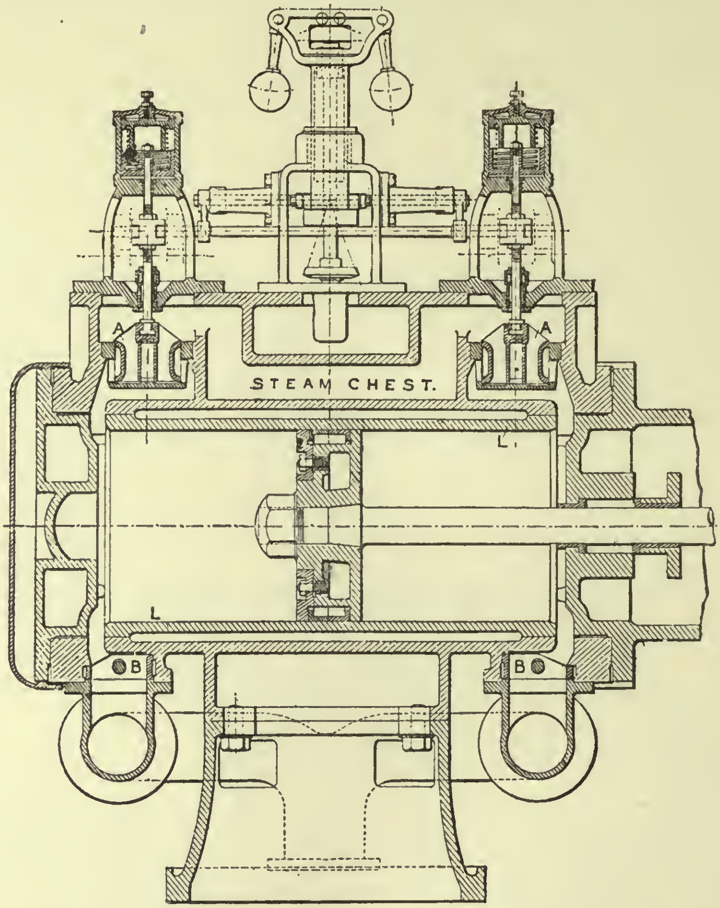


Fig. 275.—Robey Drop Valve Gear. Longitudinal Section.

While the valves are alternately raised by the engine, the point at which they are dropped depends upon the position of the governor, which regulates the cut-off at any point from nothing to $\frac{5}{8}$ of the stroke.

BB show the exhaust valves. These are triple ported, so as to give a wide opening with a small travel, and work with a minimum of friction. Being placed directly under the cylinder, this latter is kept perfectly drained ; and both admission and exhaust valve being close to cylinder body, the loss of steam in ports is avoided.

Referring now to the cross section of the cylinder, which shows very clearly the whole of the gear from one end of the cylinder, it will be seen that the small eccentric rod K is enabled to act upon the valve A (which is in equilibrium) by depressing the outer end of the lever B, by which the valve is raised.

This occurs just before the commencement of the stroke.

Owing to the different arcs described by the end of the eccentric rod and the lever B respectively, at a certain point the tripper L slips out of contact, and the valve drops instantaneously, cutting off the steam supply, as in the Proell gear.

The arrangement for securing an automatic cut-off is simplicity itself: the governor in rising moves the lever arm, and with it the pivot or fulcrum R of lever B, which of course has the effect of causing the tripper L to lose its contact, and the valve to fall at an earlier period in the stroke; and thus as the governor rises and falls the point of cut-off is accelerated or retarded accordingly.

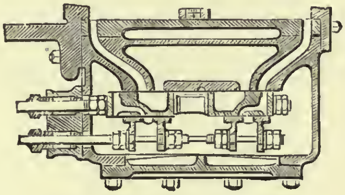
It will be noticed that the long horizontal arm of lever is prolonged past the governor, and that a small cord or string is attached to it. This cord may be led away to any part of the building, and forms a ready means of instantly stopping the engine in case of emergency, as in the case of accident to life or machinery. By pulling the cord, which is done with no more exertion than is required to ring an ordinary house bell, the lever G draws the valve levers B completely clear of the trippers, when, of course, no steam can enter the cylinders, and the engine, which may have been exerting 1000 horse-power, may be brought to a stand at once by the touch of a child.

With so sensitive an apparatus as this accurate governing is easy of attainment, and a small and simple centrifugal governor fulfils every requirement.

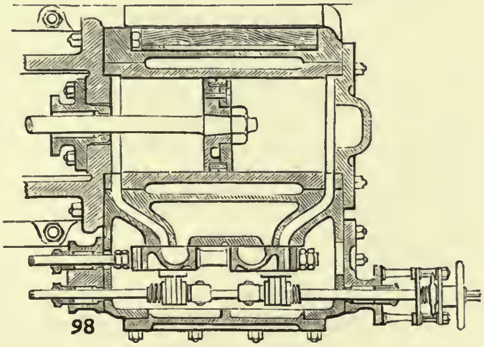
RUSTON-PROCTOR ENGINES

The first one is a long-stroke horizontal expansive working engine shown in Fig. 277.

The Hand Expansion Gear.—The slide valves are divided and placed at each end of the steam chest, so that the steam ports are very short; the expansion valves vary the period of admission of the steam from one-tenth to six-tenths of the stroke, by means of brass nuts and bushes on the spindle, which have respectively right and left-hand threads; by turning the hand wheel at the end of the spindle the valves are caused to approach or recede, cutting off steam later or earlier as may be required. An index is provided, so that the period of admission of steam can be adjusted as required.



Automatic Expansion Valve.



Hand Expansion Valve.

FIG. 276.

The illustration (Fig. 276) shows the automatic and hand expansion gear. The cut-off valve is double ported, and in the larger sizes treble ported; it is driven by a radius rod, the fore end of which is jointed to a die movable up and down in the curved slot link. The latter is oscillated by a separate eccentric, the position of the die being determined by the governor, which acts upon it by means of the lever shown. As the speed increases the governor balls rise, raising the lever and bringing the die nearer to the centre on which the slot link rocks, thus reducing the travel of the expansion valve and causing it to cut off steam earlier; should the engine run more slowly the reverse takes place, the die is lowered in the slot link, lengthening the travel of the expansion valve and prolonging the admission of the steam. The speed may be adjusted, and the oil cylinder steadies the motion. The range of expansion is from a fraction up to six-tenths of the stroke.

RUSTON, PROCTOR, & CO.'S CORLISS VALVE GEAR

A cylinder fitted with this gearing is shown in Fig. 278; the governor and rocking levers are carried by one facing, and the usual wrist plates are entirely dispensed with. The steam valve levers are mounted loose on the bonnets, and connected to the coupling rod, a trip jaw being pivoted at the outer end of each steam lever. The upper end of the trip jaw is provided with a hardened steel segment, and the lower end with a hardened steel piece. When this segment meets a roller carried by the governor trip lever (which is also carried on the valve bonnet), tripping takes place.

Each steam valve is connected direct to a vertical spring dashpot by means of a rod, the dashpots being arranged below the floor. Each dashpot lever engages with a steel piece in the trip jaw, the steam valve opening until this is tripped, which happens soon after the segment meets the roller carried by the governor trip lever. The steam valves are therefore tripped, and closed earlier or later in the stroke, according to the position of the governor.

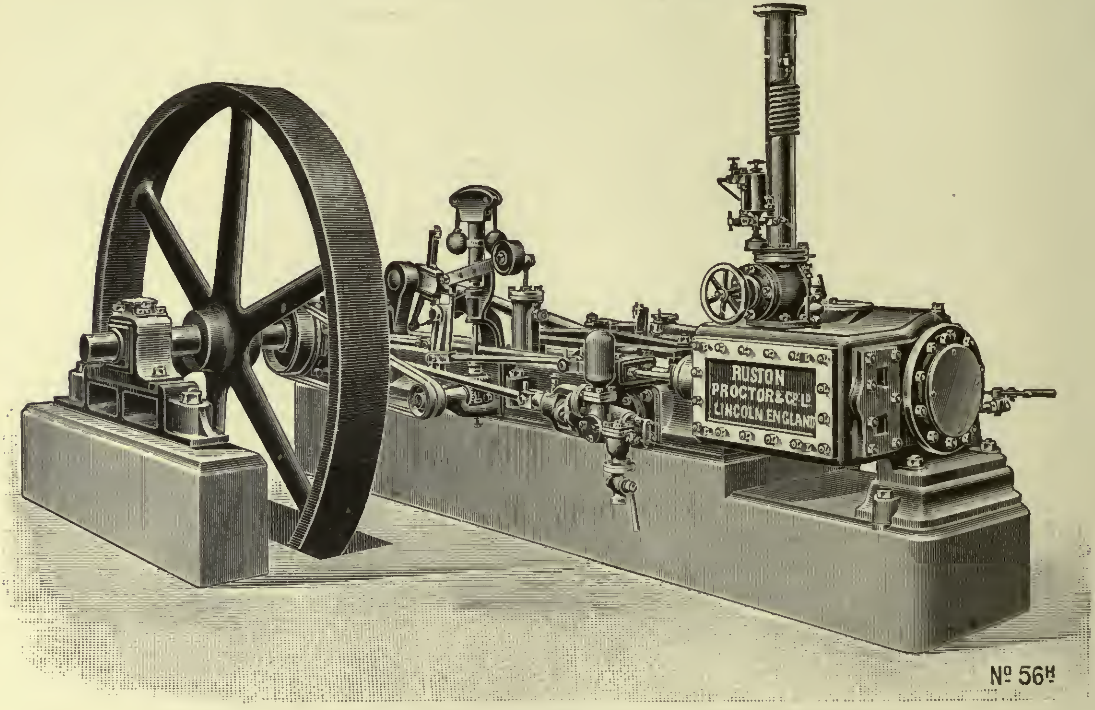


FIG. 277.—Simple Slow-Speed Engine.

The exhaust valves are positively driven by means of levers.

The system of four valves, two on the top for the admission and two below for the exhaust, enables the "clearance" to be reduced to a minimum, and the cylinder to be drained completely. By this provision also of separate steam and exhaust ports condensation of the entering high-pressure steam is to a great extent prevented, which recent investigation proves to be of the utmost importance.

All the passages being of ample area, the pressure of steam obtained in the cylinder approximates very nearly to that in the boiler; and as the same amount of coal evaporates practically the same weight of water, whether the pressure in the boiler is high or low, every pound gained in the cylinder proportionately to the boiler pressure is of consequence.

The saving to be accomplished will be realised when the loss in ordinary slide valve engines between the boiler and the initial cylinder pressure is considered, amounting to from 5 to 10 per cent., and more in economical effect, on account of the "wire-drawing" produced by insufficient valve opening. In a Corliss engine each valve has the shortest possible length of port, and, being almost in equilibrium, the power required to actuate the four valves is considerably less than for the single slide valve of a similar size of engine ; as the motion also is axial, there is every disposition for easy running.

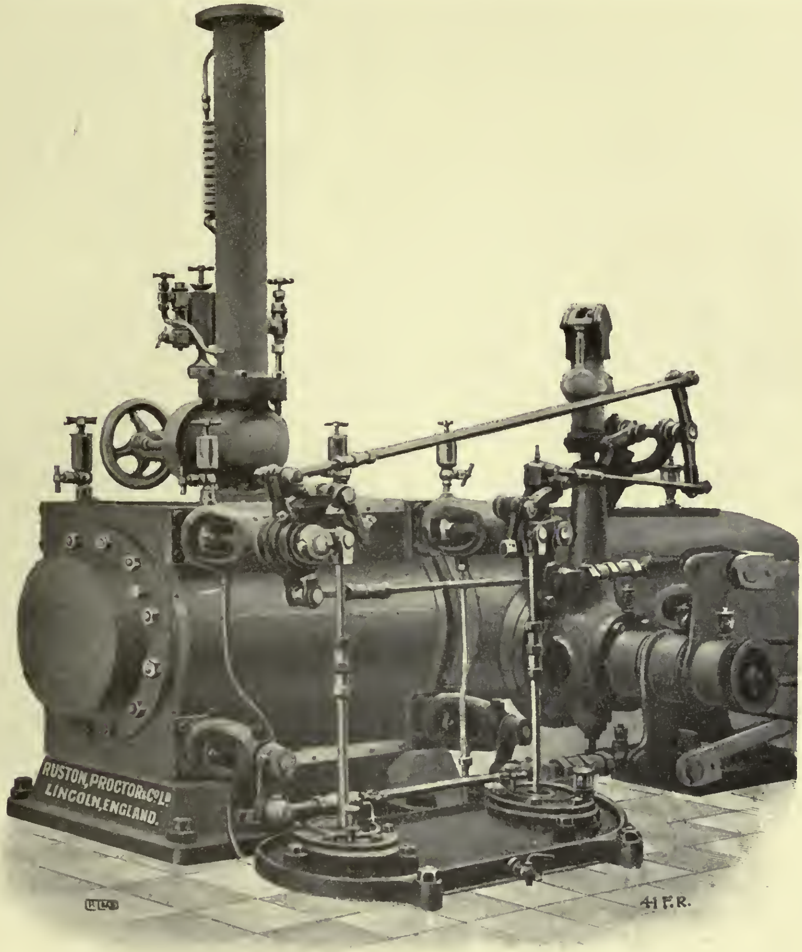

RUSTON,PROCTOR&C°.LD
LINCOLN,ENGLAND.
41 F.R.

FIG. 278.—Corliss Gear.

The requisite movement is accomplished by means of eccentrics driving a pair of rocking levers, one for the steam and one for the exhaust valve. These are arranged in such a manner as to obtain a full passage for the steam and smart closing.

The admission valves are opened as described above, but are closed by spring dashpots, the point at which each valve becomes detached from its driving lever being determined by the governor. The exhaust valves are driven positively, namely, without the intervention of trip gear.

According to the load upon the engine, the steam is thus cut off earlier or later in the stroke in a perfectly automatic manner, and the initial pressure is practically kept up until the port is closed, the valve moving almost instantaneously owing to the stored up energy in the spring; and being finally brought to rest by a cushion of air compressed in an air cylinder, which prevents the shock and vibration the spring would otherwise produce.

A merely nominal duty is thus thrown upon the governor, which has only to decide the precise moment at which the valves shall close, and not to move them, as is the case with all ordinary forms of expansion gear; a most important advantage, as regularity in running depends upon the power of the governor to quickly adapt itself to change of speed, and to impose upon the latter a heavy load, comparatively speaking, subject further to much variation, affects its sensitiveness often to a serious extent.

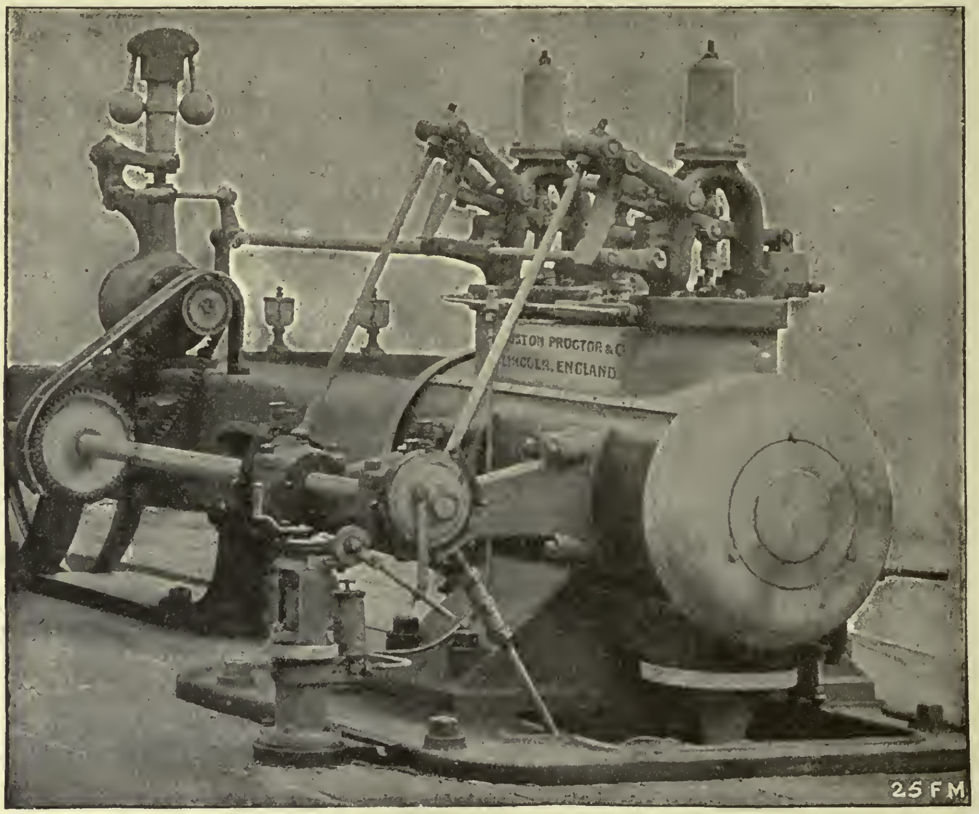


Fig. 279.—Ruston-Proctor Drop Valve Gear.

Corliss valve gear, in short, offers not only perfect economy in steam consumption, but also great regularity in speed, and engines fitted with it are well adapted for electric light work.

DROP VALVE ENGINE

The illustration (Fig. 279) shows the arrangement of the trip gear and governor, the chain drive for the latter, and the mechanical oiler for the cylinder.

The admission valves are lifted by means of levers rolling on curved steel blocks, which, when released by the trips, are forced down by springs placed above them.

The special feature consists in the particular manner in which the lifting levers are applied, and in our placing the dashpots or air buffers not immediately over the valves, but at the opposite end of the levers to which the driving rods are coupled, consequently the dashpot piston has a very large movement compared with the lift of the valve; but this is so graduated that, when the valve is only lifting slightly, the lift of the piston is from five to six times greater than that of the valve, and consequently at the very earliest "cut-off" an efficient air cushion is always available. As the valve rises the ratio between the lift of this piston and the lift of the valve decreases. At the proper point an air inlet is uncovered, so that the rest of the lift is in equilibrium, and consequently the power required is practically nil.

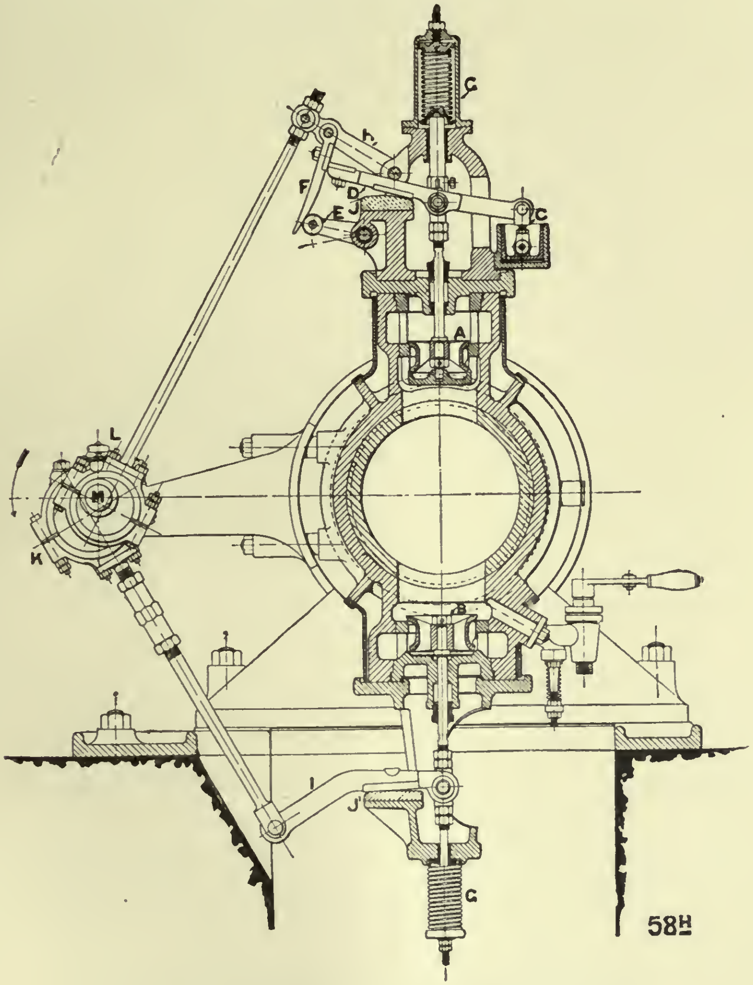


FIG. 280.—Ruston-Proctor Drop Valve Gear. Section.

This arrangement of dashpot has the further advantage that the usual stuffing box for the valve rod is not required, so that there is one less moving part on each valve to keep air-tight. It is also in a very convenient position for adjustment, and tends to keep the pressure on the various pins of the gear always in the same direction. It further enables the rolling lever motion with its graduated leverage to be successfully applied.

As just stated, the increased lift in the air dashpot piston is of no disadvantage on account of its passing the air opening at about one-third of its stroke. After the valve lever is tripped it descends in equilibrium in the same manner, until this air opening is closed, and then the cushion commences. We also claim that this position of the dashpot has a further advantage, in being independent of the line of movement of the valve spindles.

The exhaust valves, as stated, are underneath the cylinder, and are operated positively by similar rolling levers ; these are closed by springs without the intervention of dashpots.

The rock shaft immediately underneath the levers passes direct to the governor stand, and the small cranks on this shaft are provided with rollers, against which the trip levers glide ; consequently the action is remarkably smooth, and on account of the exceedingly favourable leverage described above, the work on the tripping edges is very light.

These tripping edges have the further advantage, that they can be turned by the attendant eight times before requiring to be renewed. All the pins are of extra size and properly hardened, and the main pin in the joint on the admission valve spindle (which is very liable to wear on account of the intermittent motion produced by the trip action) is provided with a special adjustment.

Fig. 280 is a sectional view of an engine cylinder fitted with drop valves and trip gear. One of the steam admission valves is shown at A, and one of the exhaust valves at B, each valve being actuated by separate eccentrics L and K working on the side shaft M.

The steam admission valves are each lifted by a lever D (riding on a curved fulcrum block J) engaging the trip latch F at one end, and connected to an air dashpot C at the other end. The lever D becomes detached as the trip latch F glides on the roller E, permitting the valve A to close instantly by the action of the spring G above it. Silent closing is obtained by an air cushion in the dashpot C. The "cut-off" takes place earlier or later in the stroke according to the position of the roller E, which is varied by the governor ; the admission of steam to the cylinder is therefore regulated in proportion to the actual load on the engine.

Each exhaust valve is operated positively by an eccentric K and lever I, also rolling on a fulcrum block.

TANDEM COMPOUND DROP VALVE CONDENSING ENGINE

For large powers this engine is shown in plan and elevation in Fig. 282. It is fitted with positive valve gear.

The following are the chief advantages of positive valve gear over trip gear :—

The steam and exhaust valves are operated mechanically, no trip gear or dashpots being employed, thereby permitting a higher speed, ensuring silent working and certainty in action with the most favourable movements.

The governing is of the most perfect character, owing to the type of governor employed and the ease with which the gear is actuated.

The piston rods are provided with a simple metallic packing ; steam joints are made "metal to metal," valve seats and covers are "ground in" ; pins are of extra large size and carefully hardened, and the valves and seats are cast from the same mixture of metal, preventing undue wear.

The engines are built for working with superheated steam (of up to 200° Fahr. above the temperature of saturated steam at the working pressure), thus effecting considerable economy.

The cylinders are steam jacketed, the steam and exhaust valve seats being cast separately ; cast-iron hoods are fitted over each of the valve chambers, and the back cover for the high-pressure cylinder has an air space.

The outer surface is lagged with blue sheet steel, and the back and false covers are polished.

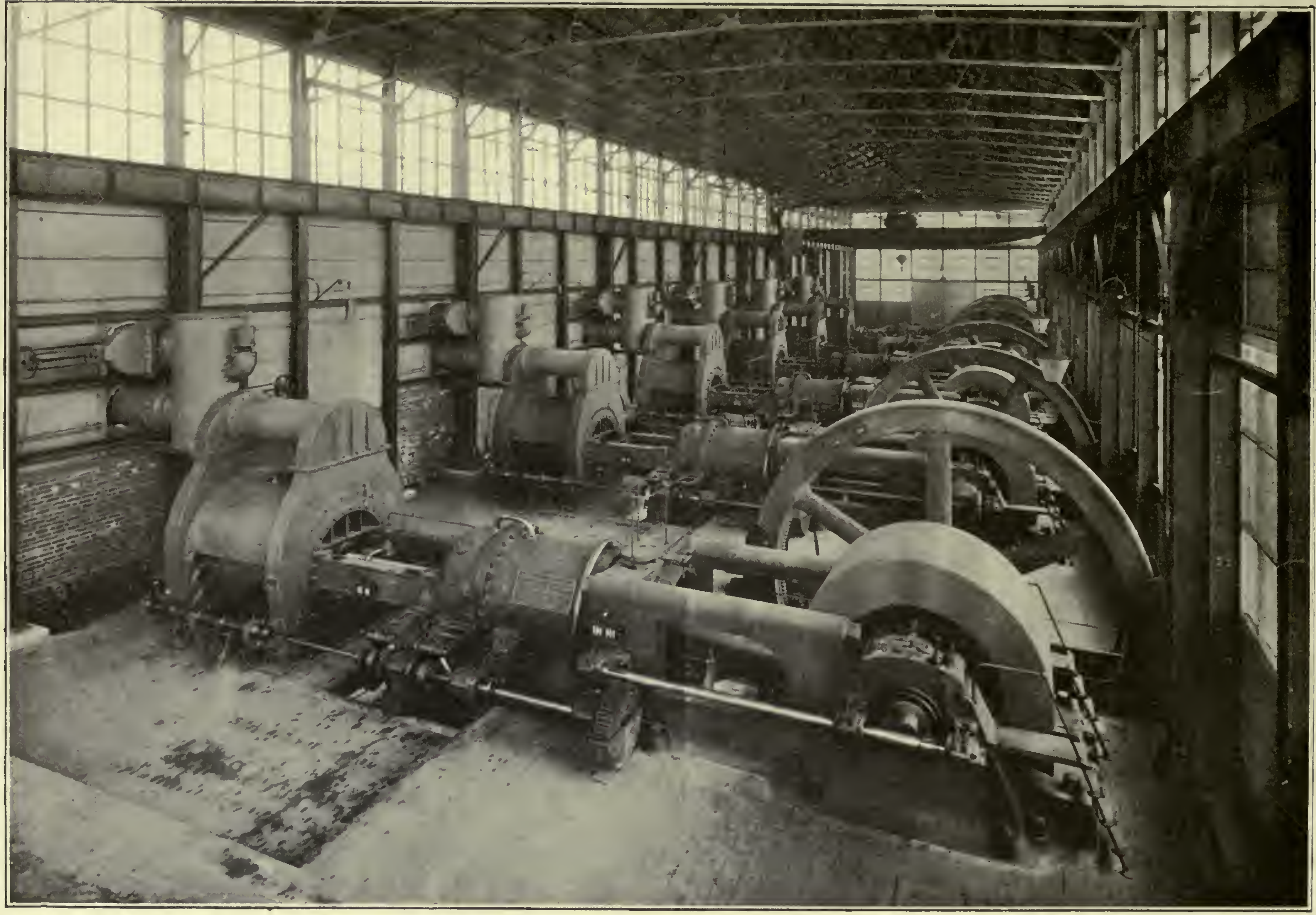


FIG. 281.—Group of Blast Furnace Blowing Engines. Driven by Blast Furnace Gas.

The cylinders are mounted on cast-iron stools bolted to a cast-iron bridge piece, which firmly ties the foundation and acts as a drip tray for oil.

The receiver is jacketed with live steam, and has a separate liner.

The stop valve is of heavy pattern, and arranged so that the spindle can be repacked whilst the engine is running. Each cylinder is provided with relief valves, drain and indicator cocks.

The four steam and exhaust valves are driven by two eccentrics, one eccentric driving a steam and exhaust valve at each end of its respective cylinder. The steam valves are on the top of the cylinder, and the exhaust valves are arranged underneath, keeping it well drained. The steam and exhaust valves are lifted by levers, rolling on curved blocks and closed by springs ; the springs for the steam valves of either cylinder are interchangeable, and likewise those for the exhaust valves.

The "cut-off" in the high-pressure cylinder is varied by the governor twisting the angle of a slide in which the free end of the eccentric rod runs, causing a pin in the centre of this rod to move in an elliptical path ; this pin actuates another lever, the free end of which is connected to the rod which drives the valve, whilst the opposite end moves in another slide also twisted by the governor, but in a reverse direction to the main slide. The effort of the governor is thus perfectly balanced, and an exceedingly advantageous movement of the valve obtained, giving quickly the proper opening and a smart closing at all degrees of expansion, and producing a steam distribution of the best and most economical character possible.

The valve gear of the low-pressure cylinder is on the same principle, but without the compound motion, the slides being regulated by hand to effect the variable "cut-off."

DANIEL ADAMSON'S VERTICAL ENGINE

On Plate VII. is illustrated a pair of inverted coupled compound engines, with both high and low-pressure cylinders supported on strong cast-iron entablatures and cast-iron box section vertical frames, on which are fitted the guides for cross-heads, the front of the engines being supported by forged steel columns fitted between base plates and entablatures. This arrangement allows of easy access to working parts for repair or adjustment. In the engines illustrated both the high and low-pressure cylinders are fitted with improved patent Wheelock expansion gear, type "B." The gear, fitted to the high-pressure cylinder being automatically controlled by the governor, whilst the gear fitted to the low-pressure cylinder is arranged to be regulated by hand whilst the engine is in motion, but may also be arranged to be automatically controlled if desired. In this type of engine, also, alternative arrangements of valves and gear can be arranged to special conditions or requirements, but the arrangement as illustrated is strongly recommended if the best and most economical results are aimed at.

This type of engine is also built of the triple and quadruple-expansion principle of the 3 or 4-cylinder type, best suited to the conditions and with plain or built-up crank shafts, and with different designs of vertical frames to suit, and can be arranged to transmit their power from fly-wheel fixed either between the cylinders or at either side of the engines, to suit circumstances ; and also with open or enclosed frames for automatic gravitation or forced system of lubrication. They are adapted for all duties where space is restricted and first cost of foundations is a consideration.

These engines can be supplied either jet or surface condensing, belt, rope, or gear driving, illustrated type "B" patent Wheelock automatic expansion gear and grid valves (Fig. 283).

This valve gear is of the single eccentric type, the valves being driven by means of levers keyed on the exhaust valve spindles, the steam valves being actuated from exhaust valve levers by means of stirrup or latch links, which are released or disengaged by means of cams or tappets at a sooner or later period by the governor. For single-expansion

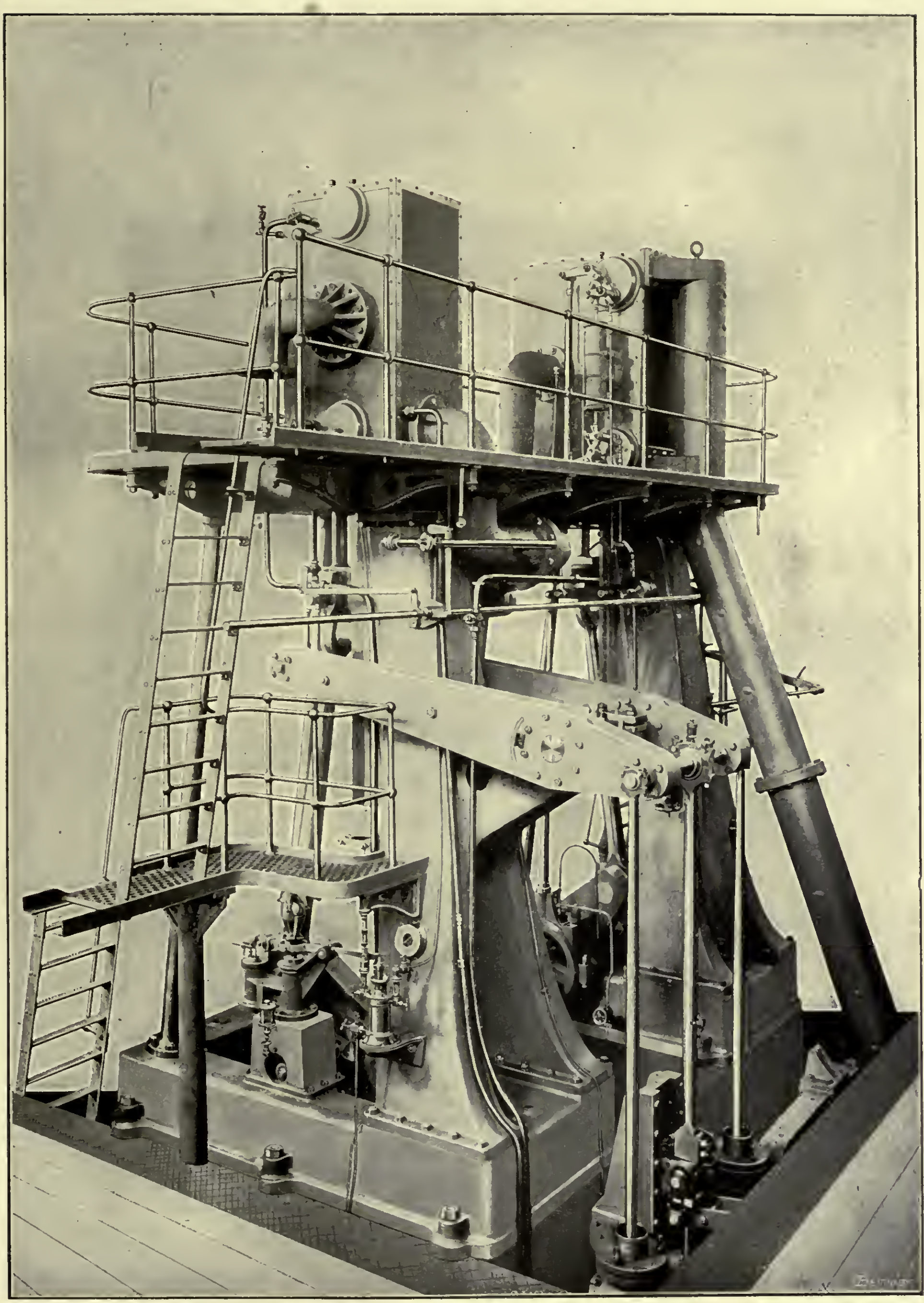


PLATE VII.—MODERN COMPOUND VERTICAL CONDENSING ENGINE FOR MILLS, FACTORIES, ETC.
(DANIEL ADAMSON, MANCHESTER).

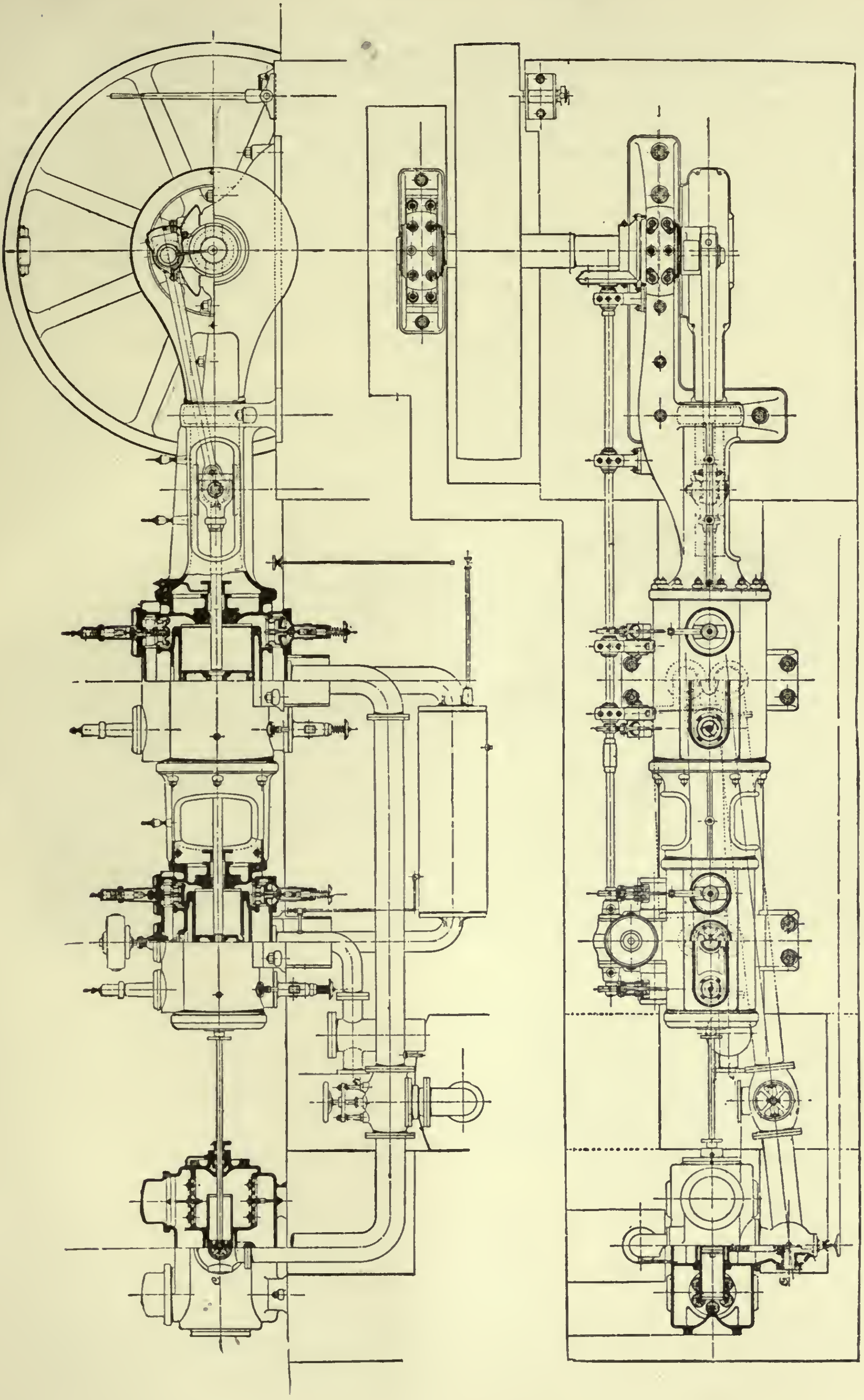


FIG 282.—Tandem Compound Condensing Drop Valve Engine.

engines the range of automatic release is up to 375 of stroke of the piston, but for multiple expansion, as illustrated, a special arrangement is provided to enable the governor to automatically control the admission of steam up to 75 of the stroke of the piston. The valve seats and valves are of the grid type, and are self-contained, being in the form of a plug (see the illustration of valve shown detached from cylinder), and is quite independent of the cylinder casting itself. These plugs are driven into suitable holes, bored a little tapering, at each end of the cylinder, no wear taking place in the cylinder, due to the action of the valves, thus allowing of spare valves and seats or plugs being kept in stock if desired. The valves and seats being of the grid type, give multiplicity of and large areas of opening, with the minimum of movement, the reciprocating movement of the valves on their seats being effected by means of knuckle or toggle joints and links, which are so arranged as to dwell or pause during the periods when the valves are closed; whilst, when opening, their movement is accelerated, giving almost instantaneous opening and closing of the valves with great ease under extreme pressures. The valve spindles are of the Wheelock patent self-packing type, entirely dispensing with the usual stuffing boxes and glands.

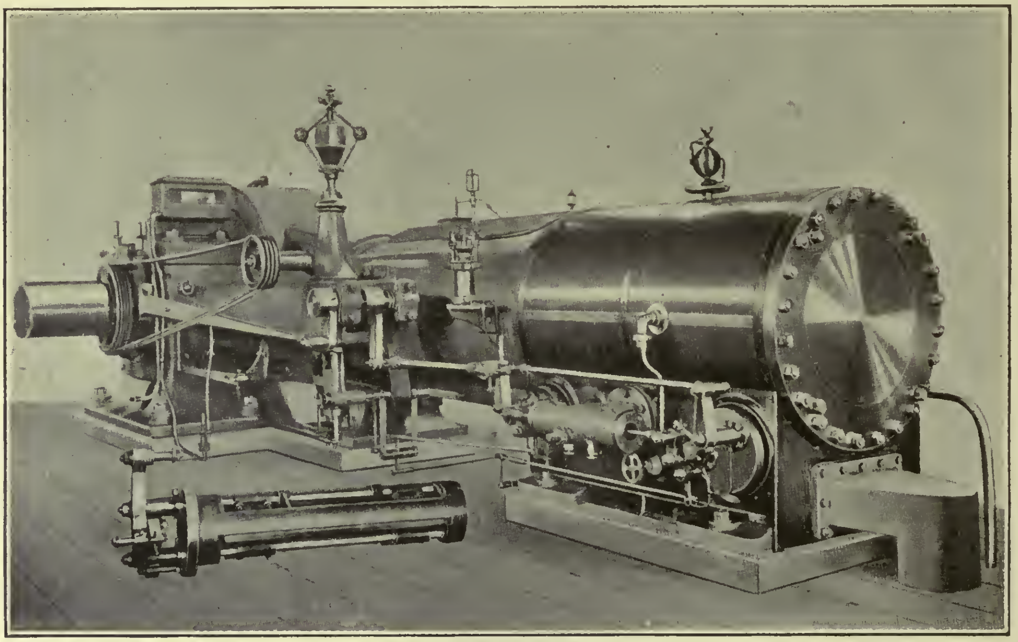


Fig. 283.—Wheelock Valve Gear.

It is simplest in construction, has the least clearance, least friction, is the most economical, and is the least trouble to the operator.

It can be applied to existing engines of any design.

COMPOUND HORIZONTAL PAIR OF ENGINES

A favourite design for mill driving is by ropes from the fly-wheel, the slow speed of the engine being multiplied by the rope drive from the large fly-wheel to smaller pulleys on the mill shafts. This system of transmitting the power will be noticed among the other systems now in vogue. The engine shown in Fig. 284 is one for this purpose. This engine is fitted with Messrs. Davey, Paxman, & Co.'s drop automatic variable cut-off valves, the latest design for which we show in Fig. 285, in section. The valves, both exhaust and steam, are double-beat equilibrium valves, with very short steam passages at each end of the cylinders.

The important features are the gears for operating the two sets of valves.

The exhaust valves are invariable, so that they are worked direct from an eccentric. The exhaust valve S has a positive motion, by means of the eccentric T and rod U, driving the fulcrum lever V, which rolls on the fulcrum block W, thereby raising and lowering the exhaust valve S from its seat.

The rolling lever V has no fixed fulcrum, so that it can slide on W, and thereby accommodate its motion to the straight-line motion of the valve spindle.

Trip Gear. — The lay shaft A is driven by means of a pair of level wheels, and rotates at the same speed as the crank shaft. Fixed on this lay shaft is the steam eccentric B, which by means of the rod C drives a pair of radius links D, which in turn carry the trip lever E. This trip lever engages with the outer end of the steam valve lever F, and raises the steam valve G from its seat, the steam valve lever being pivoted on the pin H. The valve lever F is released when the trip lever E makes contact with the roller I, which is carried by the levers K, permitting the valve G to close instantly by the action of the spring L in the dashpot M. The speed at which the steam valve closes may be accelerated or retarded by regulating the emission valve N, screwed in the side of dashpot. This air valve is so arranged that the maximum closing velocity is attained with a minimum of shock. The position of the roller I is varied by the governor, the lever O, acting upon the rod P, moving the lever Q, on the rock shaft R, on which the levers K are fixed. Therefore, according to the

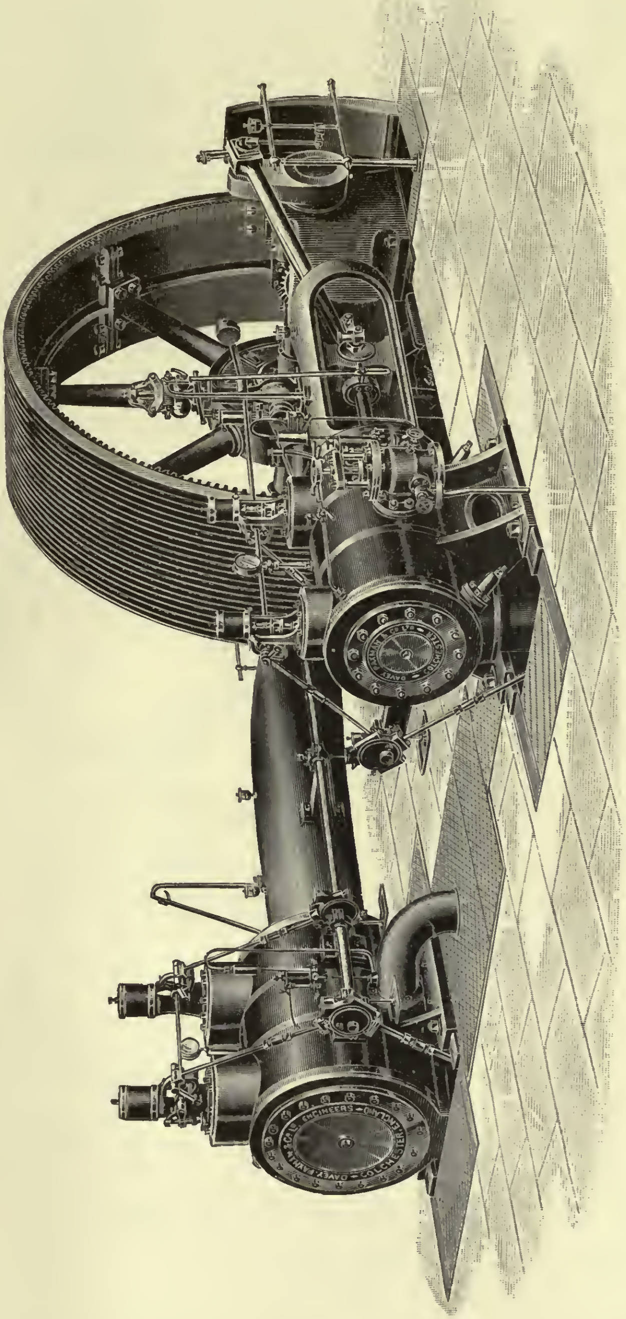


FIG. 284. — Rope Driving Engine.

position of the governor, the valve lever F is released earlier or later, as may be necessary. To obtain more or less movement to the roller I, the pin in the governor

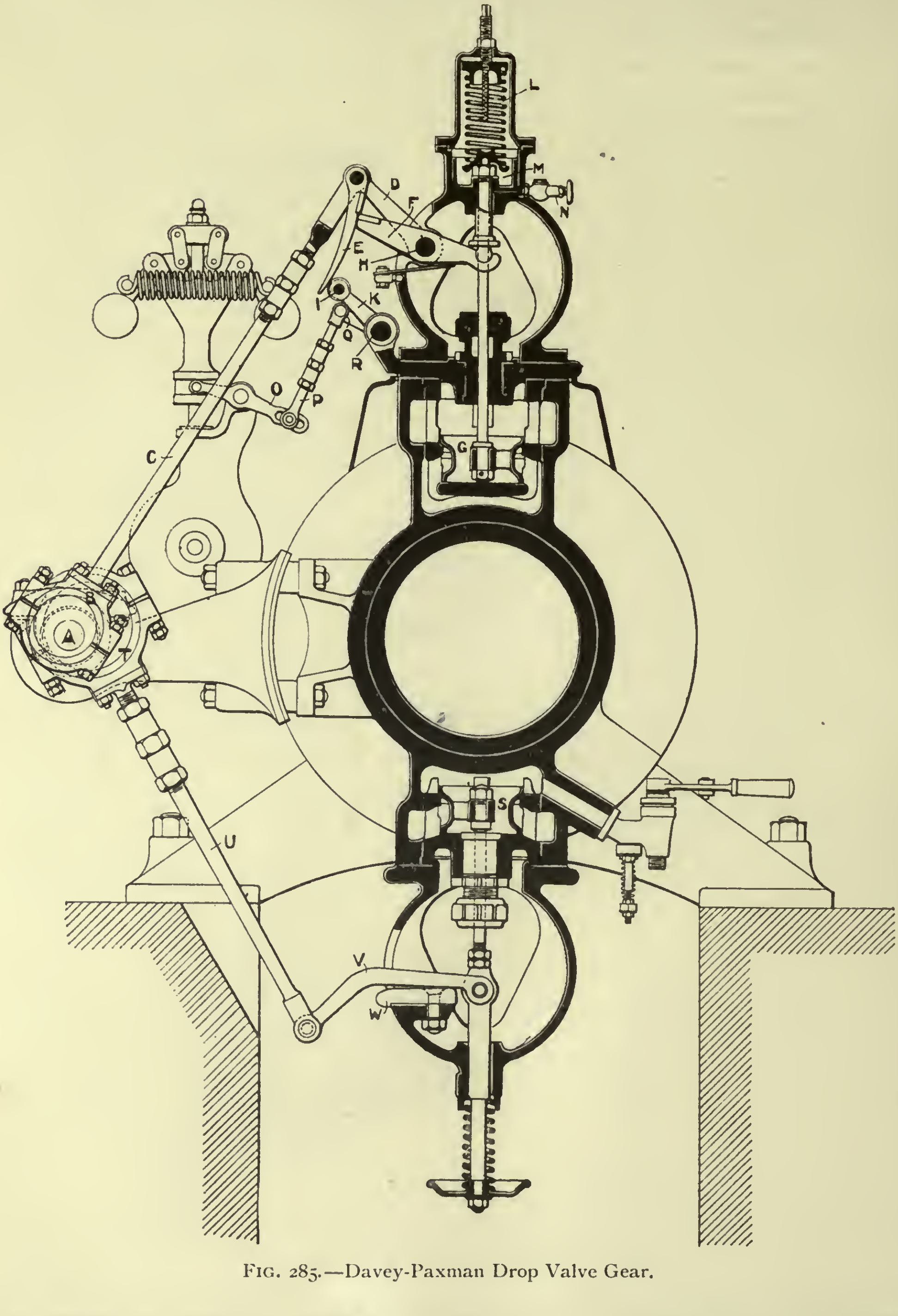


FIG. 285.—Davey-Paxman Drop Valve Gear.

rod P should be moved along the slot in the lever O. Fig. 285 shows the general arrangement of the whole gearing.

On the question of mill and factory engines it may be pointed out that, so long as only rope or belt transmission existed there was little choice of types. The old-fashioned engineers preferred the slow-speed massive engine, under the mistaken notion that it was more reliable than the higher speeded engines. No doubt their notions had some foundation, based upon the results of designing high-speed engines on the same lines as slow-speed engines, not recognising the fact that the two require different treatment.

There was also a strong prejudice in favour of horizontal engines ; only two reasons for this have been assumed to exist—first, the engine could be built on a slight frame and stiffened by binding bolts to massive masonry foundations ; and secondly, the engine looked cleaner, as all the grease, oil, and escaped water fell below, and being out of sight was out of mind. Some daring engineer, in order to get more out of an engine than his rivals, however, increased the speed, so that even the masonry could not prevent vibration. And if it was made heavy enough to prevent vibration, the strains were still there, and something broke down sooner or later. In order to remedy this, the banjo frame was introduced, in which the cylinder and crank shaft are held with the rigid cast-iron frame between. Having got so far it was evident that, given a well designed framework by which all the strains of any magnitude were provided for internally, instead of relying upon stone and bricks, an engine could be built to run in any position, and that without any great foundations.

Then came the vertical engine of the marine type, which of necessity had to be self-contained in its construction to resist the internal strains. The help of the mason and bricklayer could not be called in in this case.

The great waste of power in textile factories is a glaring abuse of the prime movers in use. The actual work performed upon cotton, silk, wool, jute, and other textile raw materials is very small. The material is soft and pliable, and the operations require little force, yet the amount of power consumed per ton of material treated is greater than that consumed by steel and iron works. This is due to the great losses in transmitting the power from the engine to the shafts, from the shafts to the machines, and from shaft to shaft and point to point in the machines.

In a Yorkshire heavy mill shed as much as 300 horse-power may be at work weaving soft threads. But the most of that power is expended in thrashing the looms, creating a tremendous noise, and more vibration than that created by a steel rolling mill. It will not always be so. This primitive state of textile engineering must give way to improvements. The smashing, crashing, pounding, and clanging of machinery handling cotton or other threads is not necessary, and is evidence of a very backward state of the art of weaving, a state which accounts for the enormous expenditure of force and energy to accomplish so little real work done.

In most modern factories the machinery employed is efficient, and therefore silent and enduring, and the engines generally fairly efficient if of any considerable size. The chief loss in these cases lies between the engine and the machinery.

At present we are concerned only with the engines. And the questions of high versus low speeds ; or, in other words, quick-revolution versus slow-revolution engines. In electrical engineering practice the conditions are such that only the best engine, no matter what its type might be, could survive. In this case the choice of engine is not controlled in any way by a textile manufacturer or other party not an expert engineer, and after the choice is made, the engine's performance is critically and constantly observed by skilled engineers assisted by most modern instruments of accuracy for tests. The sole object of every one concerned is to obtain an engine to give the greatest brake horse-power per pound of coal consumed, with the lowest cost for repairs and maintenance.

And these are exactly the two desiderata of every power user as well as the electrical engineer. It is safe, then, to take the result of the electrical engineer's experience as a guide, and this points emphatically to the quick-revolution engine.

The slow-speed engine has all the ancient experience in its favour, and its merits and demerits are too well known to require rehearsal here. But the views of the latter-day engineer, and his reasons for preferring quick-revolution engines, are worth giving fully, and we give the views of Mr. Crawford, as representing Messrs. Belliss & Morcom's engines of high-speed and forced lubrication fame.

It is usual to describe a quick-revolution engine as a "high-speed engine," and a slow-revolution engine as a "slow-speed," the only elements taken into consideration being speed of revolution and, sometimes, piston speed; but piston speed clearly affects the wear of the cylinder rubbing surfaces, piston rods and motion bars only, and as the radial pressure of a piston packing ring should be as small as possible—namely, the ring should be merely floating—and as piston rod glands should be kept tight by capillarity only, and as motion bars can always have practically unlimited surfaces, the wear on each is infinitesimal, and is not appreciably affected by even great variations in piston speed; moreover, increased piston speed can be met by increased surfaces. Piston speed need therefore only be considered in so far as it causes knock on the crank pins and main bearings. Further, as speed of revolution does not in itself cause wear—this being caused by speed of revolution multiplied by the circumference of and pressure on the revolving shaft, namely, speed of passing surfaces multiplied by the pressure on them—the main elements are the work due to impact and the friction on the main bearings, which have to be dissipated in the form of heat.

In the table on page 207 is given a comparison of 2000 brake horse-power engines of different types, which illustrates the relative values of piston speeds, speed of revolution, and speed of passing surfaces, etc.

An inspection of this table shows that the so-called "moderate" speed engine is the highest speed engine, and that the single-acting engine is a 30 per cent. higher speed engine than the Belliss double-acting, and there is very little difference between the slow revolution engine and the quick revolution engine, hitherto differentiated as "high" and "slow" speed, all of which points to the probability that some of the terms hitherto used to describe speed of engines were designedly catch terms. The table also illustrates what bad engineering it is to put the generator in between the cranks, and independently of what is shown by the numbers on page 207. The relative figures do not give the amount of energy that has to be dissipated by the bearings in each case, but, taking into consideration the constant thrust on the moderate speed engine due to the excessive, though necessary, weight of the generator, fly-wheel, and shaft, the bearings of that engine, though under forced lubrication, would have to dissipate more energy in a given time than in an engine of the Belliss type, other things being equal. Further, taking into consideration the amount of total friction in the slow-speed revolution engine, due to the absence of efficient lubrication, the heating of the bearings, as is proved in practice, is much greater; and moreover, many so-called slow-speed engines that ten years ago were designed to run at eighty revolutions are now, without any alterations in their proportions, running at 140 revolutions, and are still called slow-speed engines.

The advantages claimed by Messrs. Belliss & Morcom are as follow:—

The first point of advantage is the reduction of space occupied by both engine and dynamo. The quick-revolution type probably occupies from one-half to one-third of the space of the slow-speed set; and when an engineer finds he can get at least double the power into his station by departing from the slow-speed and adopting quick-revolution sets, although he may have a prepossession in favour of the slow-speed, he often succumbs to the logic of facts, and goes in for high-speed sets. With these he finds that there is a big reduction in first cost, less ground space to pay rent for, smaller station buildings, and less expensive travelling cranes to provide, together with many other advantages.

Quick-revolution sets being self-lubricating, there is no running about with oil cans, careful adjustment of oil syphons, supervision of numerous oil pipes to see that every bearing is getting its proper quantity of oil, feeling the bearings at frequent intervals, etc., as is necessary with the slow-speed engine. This naturally results in a reduced cost

COMPARISON TABLE OF HIGH-SPEED *VERSUS* LOW-SPEED ENGINES, BY MR. W. CRAWFORD.

Description.	Belliss 3-crank Triple.	3-Crank Single-Acting Triple.	2-Crank Compound "Moderate Speed"?! with Generator between Cranks.	3-Crank Triple Open Marine Type.	2-Crank Compound Open Marine Type.
Cylinders—ins.	23, 35, 53	18.8, 30.3, 48	33, 62	25, 37, 61	32½, 66
Stroke—ins.	30	23.6	30	48	54
Revolutions	200	200	150	85	83
Piston speed	1000	787	750	680	747
Length of engine	21 ft. 3 ins.	22 ft. 6 in.	22 ft. 6 in.	24 ft.	24 ft. 6 in.
Breadth	8 ft.	11 ft. 2 in.	21 ft. 6 in.	15 ft.	18 ft.
Height	22 ft.	30 ft. 6 in.	21 ft.	24 ft.	28 ft.
Floor space—square ft.	170	251	484 (Including alternator)	360	441
Weight of H. P. piston—lbs.	1 off 1110	3 off 118	1 off 1050	1 off 1080	1 off 2240
Weight of I. P. piston—lbs.	1 off 1078	3 off 280	...	1 off 1980	...
Weight of L. P. piston—lbs.	1 off 1089	3 off 830	1 off 2760	1 off 2800	1 off 8960
Weight of piston rings—lbs.	...	3 sets 159	...	...	...
Weight of piston rod—lbs. Weight of piston rod cross-head—lbs.	3 off 1666	3 off 1488 3 off 2978	2 off 2450	3 off 2072	2 off 3920
Weight of connecting rod—lbs.	3 off 1485	6 off 1500	2 off 2940	3 off 3584	2 off 8064
Weight of piston lines, total—lbs.	12,730	26,559	14,590	22,828	35,168
Weight of piston lines, each—average	4243	8853	7295	7609	17,584
Weight of piston lines—per I. H. P.—lbs.	6.07	12.6	6.95	10.9	16.7
Weight of crank shaft	5 tons 8 cwt.	12 tons	51 tons 8 cwt. (Including alternator)	14 tons 2 cwt.	9 tons
Weight of alternator shaft	...	...	...	14,804 lbs.	7 tons 10 cwt.
Weight of fly-wheel	...	...	32 tons	...	...
Weight of alternator	...	...	Included above	...	82 tons 10 cwt.
Weight of alternator, alternator, shaft and fly-wheel, taken by engine—lbs.	...	...	115,000	...	...
Total revolving weight, taken by engine bearings—lbs.	12,096	26,880	115,000	31,584	20,160
Total pressure on lines due to steam pressure—lbs. $= \frac{\text{I. H. P.} \times 33{,}000}{\text{Piston speed}}$	69,300	88,000	92,500	102,000	93,000
Total load on crank pins due to steam + weight of piston lines—lbs.	82,030	114,559	107,090	124,828	128,168
Total load on engine bearings due to steam + weight of piston lines + weight of revolving parts—lbs.	94,126	141,439	222,090	156,412	148,328
Main bearing, total surface—square ins.	1212.5	1790	1296	1350	1280
Main bearing, pressure per square inch—lbs.	77.5	79	170	116	116
Main bearing, rubbing speed × pressure	44,600	60,100	120,000	38,600	40,400
Crank pin bearings, total surface—square ins.	495	808	322	675	320
Crank pin bearings, pressure per square inch—lbs.	166	142	333	185	400
Crank pin, rubbing speed × pressure	95,600	108,000	150,000	61,500	139,000
System of lubrication	Forced	Splash	Forced	Ordinary	Ordinary

for attendance. It is not unusual to find one man in charge of five or six engines, but no one would think of running slow-speed sets of the same power with less than one man to each pair of engines, if not to each engine.

Facility for Overhaul and Adjustment.—The parts of the quick-revolution engine are much less massive and cumbersome than the similar parts of the slow-speed engine, and many parts, which in the quick-revolution can be man-handled, will require with the slow-speed engine a crane and men to work it before they can be moved about.

Oil Consumption.—This again is in favour of the enclosed quick-revolution type. The same oil is used over and over again, with the result that the consumption is reduced to a figure which is surprisingly low, and not only does the slow-speed or semi-high-speed open engine use a great deal more oil, but the oil is thrown about the engine-room, with consequent discomfort and mess.

The quick-revolution enclosed engine, oiled as it is on the forced lubrication principle, is naturally more efficient than its more massive slow-speed rival, and especially at the lighter loads. The chief factor in determining the efficiency of an engine is the power required to work the engine itself. As a rule the guarantees given with slow-speed engines are based on water consumption per indicated horse-power, which takes no account of the efficiency. The proper basis of comparison is on the actual output, namely, water per brake horse-power, and on this basis it is often found that an engine which has a considerable advantage in steam consumption per indicated horse-power is really not the most economical engine. The comparison is also often made at the full load only, ignoring the fact that owing to the exigencies of station management the engines may be mostly worked at three-quarter or even half-load, and at these low loads the economy of the slow-speed engine turns out badly, on account of the large proportion of power lost in friction. Under similar conditions, as to steam pressure and vacuum, the economical performance of quick-revolution engines per brake horse-power is at least equal to that of any slow-speed sets of the same power.

Even Turning.—Another point of great importance, especially when alternators are to be run in parallel, is uniform angular velocity throughout the revolution. With slow-speed engines this is difficult of attainment, and the fly-wheel, which on account of the slow speed has to be large, has to be made still larger to meet this requirement as to even turning, and, notwithstanding this, grave difficulties have been found in some slow-speed stations in parallel running. With a high speed of revolution this point of even turning ceases to be a difficulty, and it is found easily possible to run in parallel alternators driven even by engines with single or opposite cranks. Very serious disasters have occurred with slow-speed engines, arising from the bursting of the enormous fly-wheels they are fitted with.

Vibration.—This is an objection raised against high speed of revolution, and years ago serious difficulties arose on this score, before the subject was properly understood. Since, however, it became known that engines with three cranks at equal angles and equal weights of reciprocating parts on each line were practically non-vibrating, the trouble has departed, and large stations with engines of this type are being run in the heart of London.

Durability is a strong point of engines with forced lubrication. Although they run at exceptionally high speeds of revolution there is absolutely no wear of the journals or brasses, as evidenced by numerous engines which have been opened up after long periods of running. With engines of the ordinary slow-speed open type the bearings, as is well known, require attention and adjustment at frequent intervals. Hot bearings, which are such a frequent source of serious trouble and annoyance with ordinary open type slow-speed engines, are practically unknown with forced lubrication. What these facts mean in saving of wear and tear and expense of overhaul and adjustment, any one who has had experience with ordinary open engines can testify.

Weight for Shipment and Transport.—Perhaps the strongest argument in favour of quick-revolution engines for export will be found in the fact that power for power the weight is only one-fifth to one-fourth of the slow-speed plant; in fact, the fly-wheel alone of the slow-speed engine will often weigh more than the whole high-speed engine. This fact is of the greatest importance when engines have to be transported long distances in newly developed countries where the means of transport are inadequate.

Some of these claims refer particularly only to electrical plants, but the main claims are applicable to all the uses of power.

There is no reason whatever to suppose that high speed in itself should lead to undue wear and tear. In the author's own experience the maintenance of large slow-speed engines working constantly fully loaded has not been less than that of high-speed engines, while the latter have had other advantages mentioned above.

Now that the antiquated system of power distribution by ropes, belts, and pulleys, and shafting has been so signally proved far inferior to electric transmission, the choice of engine for mill and factory driving is governed more from the electric generator point of view, and that leads at once to high-speed engines direct coupled to the generator. And if condensing water is available in plenty, the turbine must be considered in making the selection. Whether others have had the same experience cannot be ascertained, but the author has found nothing whatever to complain of in turbine installations in connection with the turbine itself, but the performance of the dynamos connected to them has never been entirely satisfactory. Neither continuous nor alternating dynamos seem to work well. This trouble may be due to the high speed of the dynamo armatures, and in connection with turbines is one to be reckoned with.

HIGH-SPEED STEAM ENGINES

Nearly every steam engine maker has brought out a high-speed engine of one or other type, single or double acting. Those who anticipated the demand for high-speed engines have reaped their just rewards for their enterprise and foresight. Those who have followed after the demand are likely to meet with the competition of turbines and gas engines to a greater extent.

The Willans engine (Fig. 286) first came into prominence, and has been very largely adopted in many electrical installations.

We have already referred to its valves and valve gearing (pages 40 and 41), two of its special features, and now describe its general construction.

The engine shown in the sectional figure (Fig. 287) is a triple-expansion engine. It will be noticed that in this engine the steam is in the cylinder, and at work during three complete revolutions of the engine. The air cushioning, another special feature, of the moving column of pistons and rods is an important necessity with single-acting engines, in order to absorb the inertia at the up-stroke end.

Reference has been made to the fact that the connecting rods, and all the moving parts are constantly in compression—a condition rendered possible only by the fact that the pistons are single acting, giving no pull to the crank upon the up-stroke, but only a push upon the down-stroke. In any engine running at high speed, however, the moving parts can only be kept in compression upon the up-stroke by very powerful cushioning, which is rarely obtained in other high-speed engines without excessive compression in the cylinders, involving wasteful use of the steam. Sometimes, when a high-speed engine exhausts into a vacuum, sufficient cushion cannot, by the usual means, be obtained *at all*. In the Willans engine very little compression is given in the steam cylinders, for little or none is required; the requisite cushioning is obtained independently by special means. It is, in fact, provided, without the addition of a single moving part to the engine, by the guide pistons. These, on the up-stroke, compress the air contained in the guide cylinders, and thus any desired amount of cushion can be obtained, according to

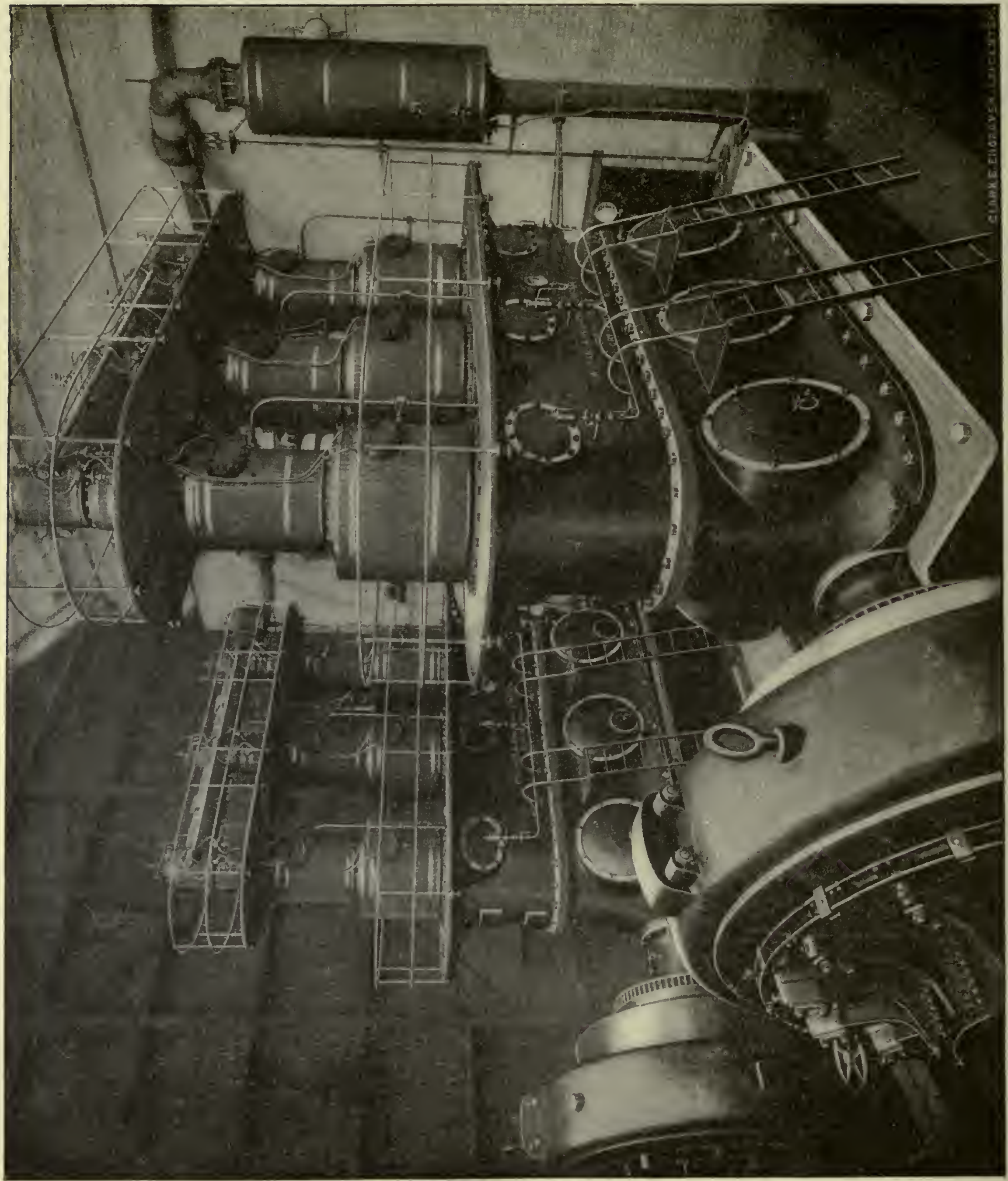
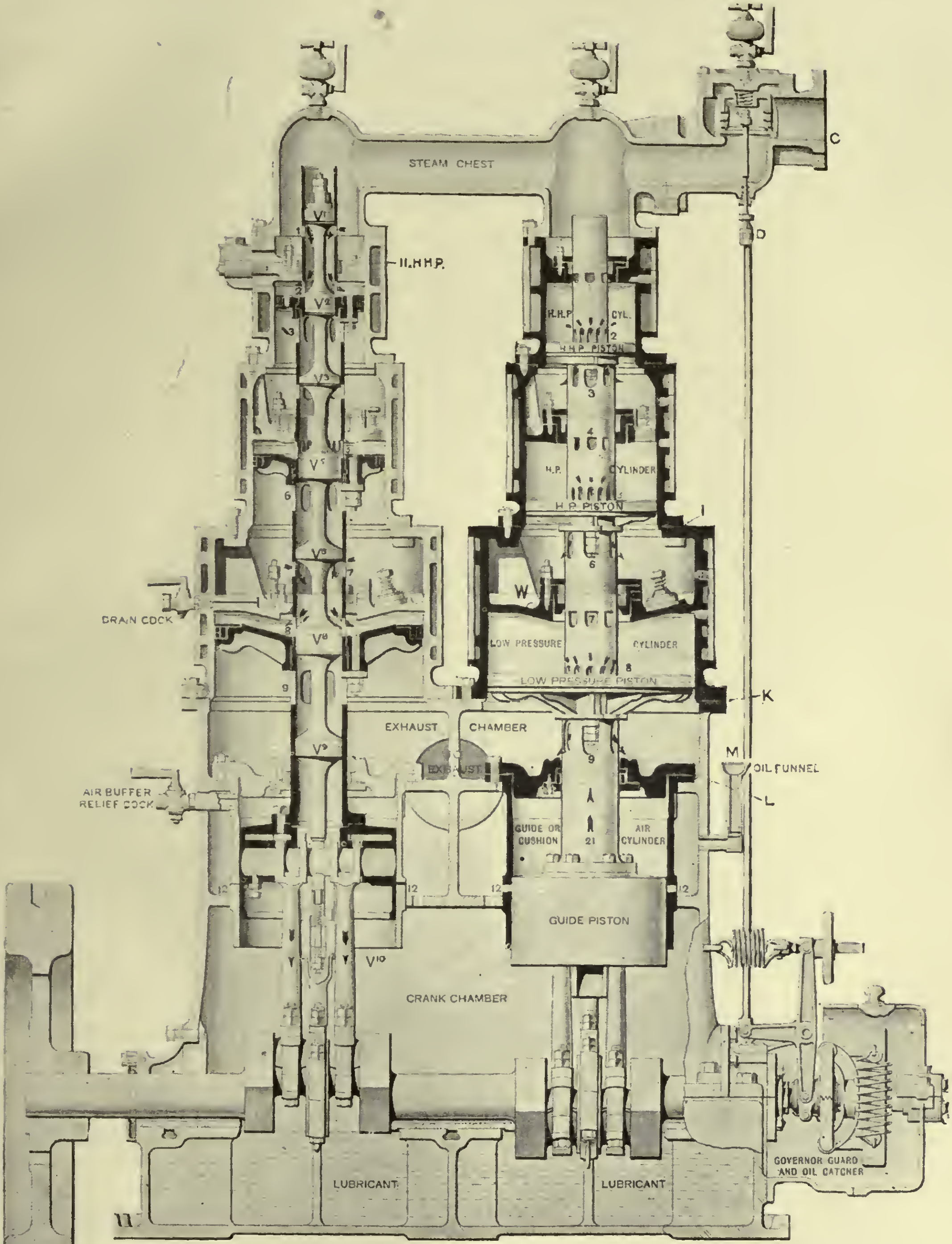


FIG. 285.—Installation of Willans' Engine.


STEAM CHEST
C
D
H.H.P.
V1
V2
V3
V5
V6
V8
V9
V10
1
2
3
6
8
9
12
H.H.P. CYL.
H.H.P. PISTON
H.P. CYLINDER
H.P. PISTON
I
W
LOW PRESSURE CYLINDER
LOW PRESSURE PISTON
K
EXHAUST CHAMBER
EXHAUST
M
OIL FUNNEL
L
GUIDE OR CUSHION AIR CYLINDER
21
GUIDE PISTON
CRANK CHAMBER
LUBRICANT
LUBRICANT
GOVERNOR GUARD AND OIL CATCHER
DRAIN COCK
AIR BUFFER RELIEF COCK

PISTONS ON DOWN STROKE; ABOUT ONE-FOURTH OF STROKE COMPLETED; STEAM ENTERING AS SHOWN BY ARROWS; PISTON-ROD IN SECTION, SHOWING VALVES IN ELEVATION.

PISTONS ON UP STROKE, ONE FOURTH OF STROKE COMPLETED; STEAM EXHAUSTING AS SHOWN BY ARROWS; PISTONS AND PISTON-ROD IN ELEVATION

FIG. 287.—Sectional View Willans' Engine.

the clearance allowed. The amount of cushioning is fixed in each case to suit the intended speed, and may be insufficient to prevent knocking if that speed is largely exceeded. If for any reason it is desired to run an engine materially faster than was originally intended, and the engine is found to knock at the increased speed, the speed must be reduced until the knocking disappears.

An alteration in the air cushioning must be made for any increase of speed.

The work expended in compressing the air is given out again by its expansion on the succeeding down-stroke, and the loss, when the engine is running at good speed, is proved by indicator diagrams to be too minute to be worth consideration. There are holes 12, 12, in the guide cylinders, which are uncovered by the guides at the bottom of the stroke. As the casing or chamber which surrounds the guide cylinders (and forms part of the framing of the engine) is open to the atmosphere, it is evident that the air compression always commences at atmospheric pressure, and is constant and invariable in its results, whatever alteration may be made in the exhaust steam pressure.

In the low-pressure cylinder of a Willans engine, and in the high-pressure cylinders, if large enough to be so treated, internal relief valves are fitted, consisting of gun-metal plug screwed into the top of the low-pressure cylinder. The plug is pierced by holes covered by a single gun-metal disc. When the disc is raised there is free communication between the cylinder and the receiver (or steam chest) above it. It is kept down under ordinary circumstances by the excess of the receiver pressure over that in the cylinder, aided by a spring. If from water in the cylinder, or any other cause, the pressure rises above that in the receiver, the valve lifts, and though the water is only passed back into the receiver, the relief is found to be far more effective than that given by ordinary external relief valves. Engines so fitted have been tested by discharging a cubic foot of water suddenly into the steam pipe; also by connecting the steam pipe with the water space of the boiler (by a $\frac{1}{2}$-inch pipe, with a difference of 80 lbs. between the pressure in the boiler and that in the steam pipe) without any injury to the engine in either case.

It will be noticed that a compound or even triple-expansion engine may be run as a simple engine without removing the upper cylinders, by merely removing the upper pistons and the piston valves corresponding with them; in fact, either the steam pistons or the valves might be left, were it not for the useless friction of the rings; the upper cylinders, in such a case, become mere extensions of the steam chest. It is sometimes a practical convenience to be able thus easily to alter a compound to a simple engine, or a triple-expansion to a compound.

TANGYE'S HIGH-SPEED ENGINE

A perspective view of the engine is given in Fig. 288. Fig. 289 is a longitudinal vertical section through the engine, while Fig. 290 is a transverse vertical section through the engine at a point very near the centre of the crank chamber. The engine, which is to give 60 kilowatts, is of the inverted 2-crank type, with the working parts entirely enclosed in an oil-tight casing, as is usual with this class. It is non-condensing, and works at a pressure of 150 lbs. per square inch, and runs at 480 revolutions per minute. The lubrication is effected by a pump worked from the low-pressure eccentric, which forces oil to all the important bearings. After being used the oil flows to a well in the engine base, from which it is pumped over again. Large openings in the casing, provided with doors having faced joints, permit the inspection, or removal, of any of the working parts.

One important object kept in view while designing this engine was the avoidance of complicated forms of valve gear, which is a matter well worth consideration in this class of engine. The steam distribution is effected by two piston equilibrium slide valves which work in renewable cast-iron liners. The high-pressure cut-off is automatic, depending on the load. It has a range from 0 to 6/10, which corresponds to only 2 per cent. change of speed.

The pistons are of the block type, with Ramsbottom's spring rings, and the piston rods are fitted with metallic packing. The cross-heads are forged solid on the rods, and the slippers are of cast iron, and are exceptionally large. They are turned to fit the bored guides.

The governor is of the crank shaft type, and operates on the combined effect of centrifugal and inertia forces. It is not embodied in the fly-wheel, as is frequently the case with such governors, but is separated from it, and enclosed in the crank chamber casing for the sake of cleanliness, and to ensure ample lubrication. It gives very close regulation between full load and no load, the settled variation being 2 per cent. An important feature also of the regulation is the rapidity with which the engine settles to the altered load. In the table annexed will be seen particulars of the fluctuations which occurred when using a fly-wheel of ordinary weight, but when driving alternators in parallel with these engines it would probably be found necessary to use heavier fly-wheels. The governor allows of a wide range of speed variation without its sensitiveness being in any way affected, the alteration being controlled by weights and not by springs; a total variation of speed of about 50 revolutions, or 10 per cent., is possible.

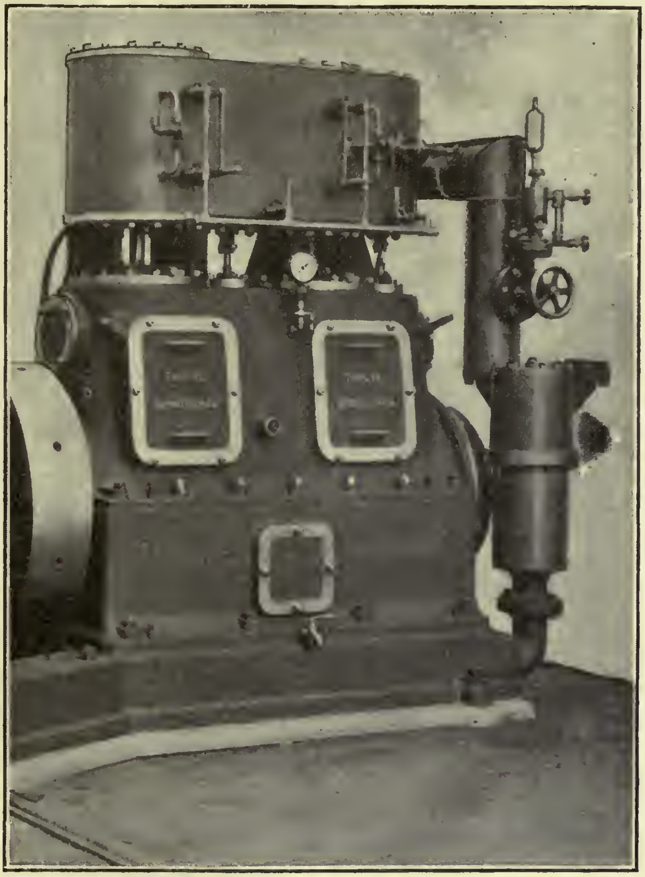


FIG. 288.—Tangye's Engine. High Speed.

The cylinders are carried on distance pieces, which are combined with the cylindrical cross-head guides, as may be seen on reference to Figs. 289 and 290 on page 214. This ensures perfect alignment of piston and connecting rod. Access to the piston rods is obtained through spaces cast in the distance pieces. The cylinders have deep steam chests with straight ports, so as to reduce clearances and also to help the drainage.

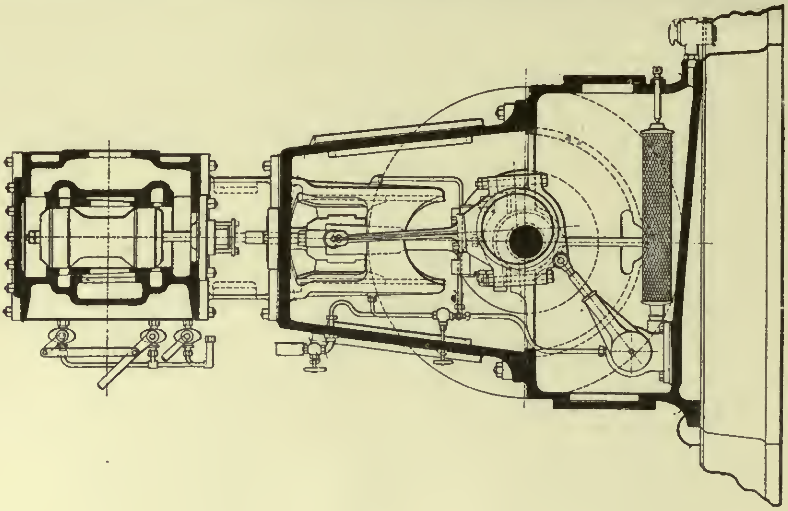


Fig. 290.—Cross Section View.

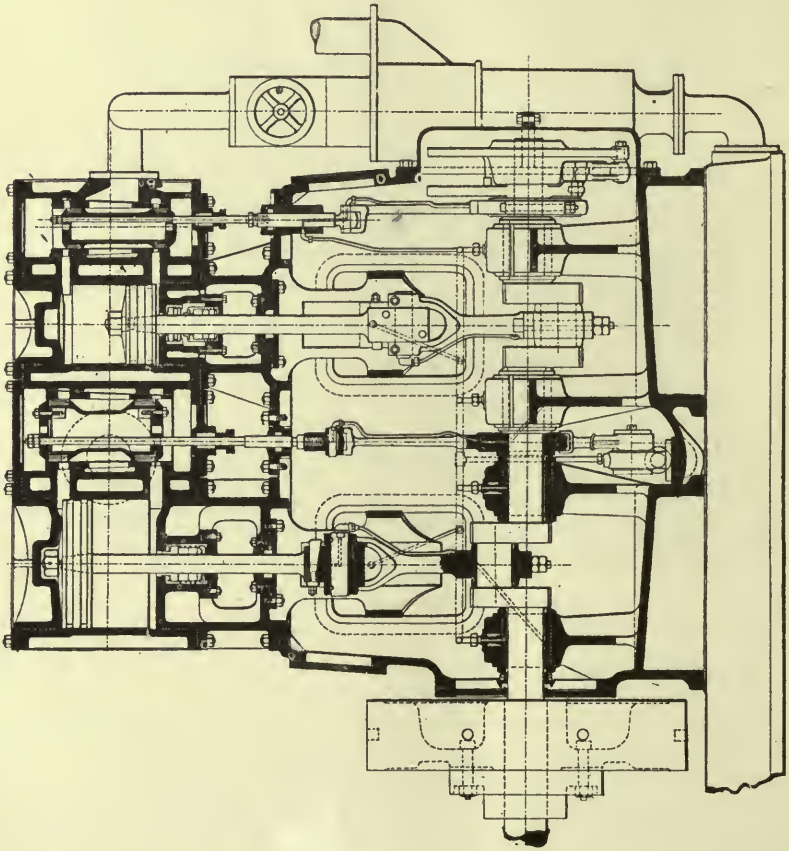


Fig. 289.—Sectional View Tangye Engine.

The engine is suitable for working in conjunction with a condenser, or with a high degree of superheat, and in the table annexed the performance of the engine at different loads is given, but the results are those obtained when using saturated steam from a water-tube boiler, and not with steam that had been subjected to any artificial drying.

The engine fittings include a combined steam trap and separator, cylinder, and steam chest drain cocks, auxiliary steam supply to low-pressure cylinder, relief valves, indicating gear, tachometer pulley, oil-pressure gauge, and regulating valve.

The engine has been thoroughly tested on a continuous current dynamo, with the following excellent results:—

RECORD OF TANGYE ENGINE AND DYNAMO TEST. ENCLOSED FORCED LUBRICATION ENGINE WITH DYNAMO NON-CONDENSING.

	Average Boiler Pressure.	Average Revolutions per Minute.	Output.		Kilowatts	Diagrams.			Pounds of Water per Hour.			Combined Efficiency.
			Average Electromotive Force.	Average Current.		Average Mean Effective Pressure.		Indicated Horse-Power.	Total.	Per Indicated Horse-Power.	Per Kilowatt.	
						High Pressure.	Low Pressure.					
	Lb.		Volts.	Ampères								
Full load .	150	479.2	248.6	240	59.6	46.8	18.3	101.9	2086	20.5	35	78.6
Half-load .	150	485	250.7	120	30	26.9	8.6	53.5	1390.6	25.9	46.3	75.7
Quarter-load	150	481	248.3	61.6	15.3	19.5	5.1	35.4	1054.6	29.7	69	58.2

Governor Trial.—Full load—240 ampères ; 250 volts ; 60 kilowatts. Running light, speed 490 ; loaded, 480. 2.06 per cent. settled variation. Highest speed reached with full load suddenly thrown off, 515. Lowest speed reached with full load suddenly put on, 450. Half-load—120 ampères ; 250 volts ; 30 kilowatts. Running light, speed 490 ; loaded, 486. .82 per cent. settled variation. Highest speed reached with half-load suddenly thrown off, 510. Lowest speed reached with half-load suddenly put on, 475 revolutions per minute.

The further consideration of high-speed steam engines will be continued in the next volume. Of the class now under consideration, only a few may be noticed. Modern practice is becoming so uniform in this line of prime movers that there is little difference to notice between them ; still, they are an important class, and represent present-day stages of development in reciprocating steam engine progress. Marine engines naturally follow for consideration, as they were, in fact, the forerunners of these high-speed vertical engines. Although for high-class high-speed vessels the steam turbine has taken the lead as the marine engine of the present and future, yet it seems that it will be some considerable time before it will supersede the ordinary marine engine for freight carrying work. The 10 or 11-knot tramp steamer is hard to beat as an economically propelled vessel, and comparatively cheap to build compared with turbine vessels.

The railway locomotive prime mover also belongs to the class of high-speed engines. It differs very little to-day from the engine of twenty or twenty-five years ago. They have long been standardised in design. The only departure from common practice has been made recently in the way of running single carriages with a steam or internal combustion engine in one end ; in fact, a railway motor car on the main lines.

There is also a movement on foot for converting the railways into electric transmission lines, and so to abolish the locomotive engine altogether. Short lines have already been set to work at Liverpool and Newcastle-on-Tyne, also underground in London, but success in these cases is far from a proof of success on long main line journeys with heavy trains. Still, there is no reason, from an engineering point of view, why the long-journey heavy train service should not be successfully run at a cheaper cost than by steam locomotives. And ten years or even less of present progress in America will in all likelihood see the steam locomotive giving way before electrically transmitted power on most important railways. As with the case of ordinary marine engines, however, it will be many a decade before the locomotive becomes a prime mover of the past; it will always have a place where it cannot be superseded. Seldom do new inventions immediately take the place of the old machinery; the displacement is gradual, and not often ever entirely completed. Candles continued to be used after gas was invented; and both are still in use although we have electric light available. The motor car and the horse vehicle will be found on the roads, probably, when the flying machines are quite familiar to engineers. It is quite safe for steam engine makers to go on with the manufacture of reciprocating engines of large dimensions and power, but at the same time the rival prime movers are not to be neglected; and especially to fresh adventures are the later developments of most interest. Additional capital is more likely to prove remunerative in promoting the manufacture of the more recent prime movers than in continuing that of the older types.

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